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THEORETICAL AND EXPERIMENTAL STUDY OF CONDENSER SCOOPS.*

By H. F. SCHMIDT, MEMBER.

Consideration of certain problems in connection with the performance of condensing equipment on Naval vessels led the writer to an examination of the designs of the scoops which have been employed for supplying circulating water through the condensers of Destroyers and Scout Cruisers. This study suggested the possibility that these scoops were not of the best form, as no means were provided to convert the velocity of approach to the scoop inlet into static pressure. This is required in order to overcome the resistance through the injection and discharge piping, condenser tubes, water boxes, etc.

* Naval Engineers' Prize Essay, 1929.

The writer was thus led to investigate the theory of scoops, and theoretical study in turn suggested the need for experimental work to determine certain design constants. The results of these studies and experiments are embodied in the ensuing discussion.

THEORY.

Scoops depend for their operation upon a well-known principle of hydraulics, namely, that if we insert a bent glass tube in a moving stream, as in Figure 1, it will be found, as shown in all works

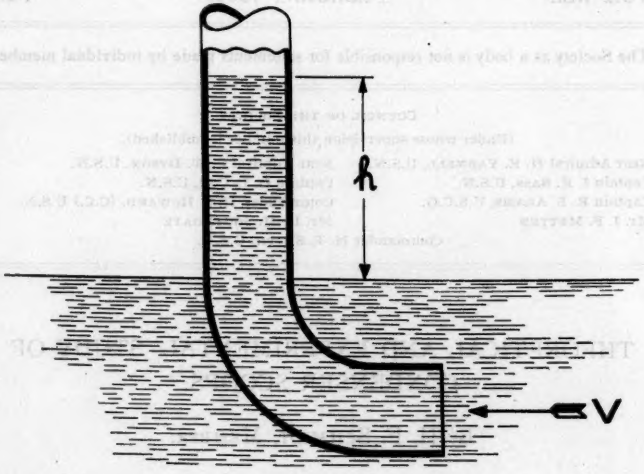


FIGURE 1.

on physics, that the water will rise in the tube to a height of h feet, as expressed by the relationship

$$h = \frac{V^2}{2g}$$

in which V is the velocity of the stream in feet per second and g is the acceleration of gravity in feet per second per second.

Even neglecting friction, it is obvious that if some of the water were permitted to escape from the tube, the height to which it rises would be reduced because the force of the impact on the end of the tube would be less than that indicated by the above formula.

We are all familiar with the Venturi Meter and the conversion of velocity into pressure by the slowing-up of the stream in the diverging portion of the tube. If V_1 is the velocity at the entrance to a diverging tube, as in Figure 2, and the inlet area is A_1, and the area at the enlarged section is A_2, then for continuity of flow, the

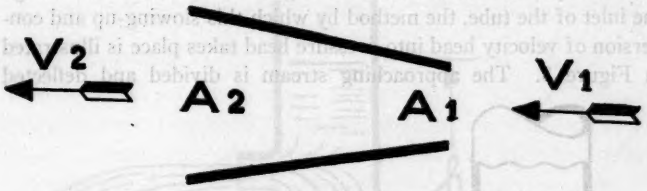


FIGURE 2.

velocity at the section A_2 will be $\frac{A_1 V_1}{A_2}$. The head built up due to the slowing-up of the liquid in a diverging tube is generally represented by the formula:

$$h = \frac{V_1^2}{2g} - \frac{V_2^2}{2g}$$

in which V_2 is the velocity at the enlarged section. Hence, for the section A_2, the head becomes:

$$h = \frac{V_1^2}{2g} - \frac{V_1^2}{2g}\left(\frac{A_1}{A_2}\right)^2$$

Now the quantity of fluid entering the small end of the diverging tube will be $V_1 A_1$, and its kinetic energy will be

$$A_1 V_1 w \frac{V_1^2}{2g}$$

where w is the weight per cubic foot. Hence, the efficiency, as a fraction of the available velocity head at the inlet which is converted into static pressure at the enlarged end of the tube, will be

$$E = \frac{w A_1 V_1 \left[\frac{V_1^2}{2g} - \frac{V_1^2}{2g}\left(\frac{A_1}{A_2}\right)^2\right]}{w A_1 V_1 \frac{V_1^2}{2g}} = 1 - \left(\frac{A_1}{A_2}\right)^2$$

That is, the static efficiency is dependent only upon the ratio of the areas of the diverging tube.

In the above theory of the diverging tube, it is assumed that the flow through the tube is such that the velocity through the inlet, or small end, is equal to the velocity of approach of the stream. As previously pointed out, it is necessary to slow-up the stream gradually in order to convert velocity head into pressure head. In the case of a parallel tube, such as in Figure 1, with no flow through the inlet of the tube, the method by which this slowing-up and conversion of velocity head into pressure head takes place is illustrated in Figure 3. The approaching stream is divided and deflected

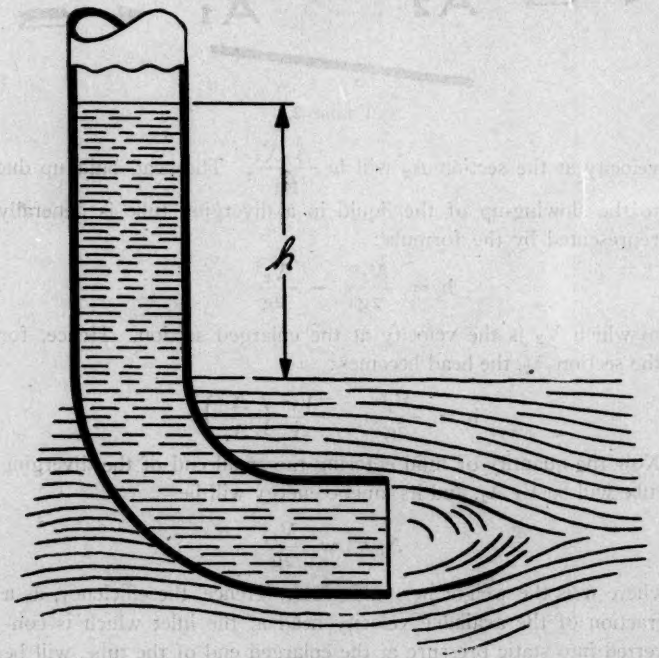


FIGURE 3.

around the end of the tube, leaving a cone-shaped mass of fluid ahead of the tube entrance. This conical tube of liquid is really in effect a diverging tube, the entrance area of which is zero and the other end the area A_1. As the ratio of areas is infinity, the velocity at the large end of the zone will be zero, and the equation for the diverging tube reduces to

$$h = \frac{V_1^2}{2g}$$

Now if, instead of the bent tube of Figure 1, we had a condition where some of the water were permitted to escape from the tube, the height to which the water would rise in the tube would, of course, be decreased, and, under these conditions, we can conceive of the flow around the inlet as being as illustrated in Figure 4, in

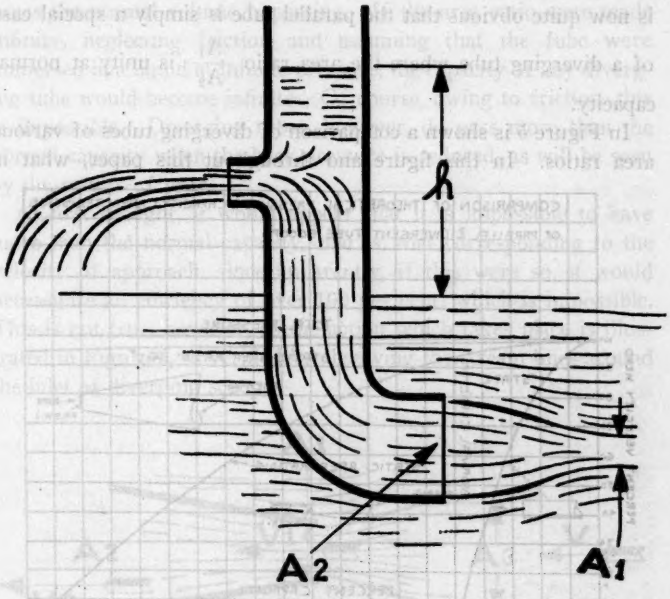


FIGURE 4.

which the central column of the approaching water is of such an area that it will pass just the amount of water at the velocity of approach as is being taken out of the tube. The area of this column now becomes A_1 in the general formula and the entrance area of the tube, A_2. Thus we see that for any condition of flow through the parallel tube, the static head available is still represented by the formula

$$h = \frac{V_1^2}{2g} - \frac{V_1^2}{2g}\left(\frac{A_1}{A_2}\right)^2$$

It is obvious that if the flow through the tube were such that the velocity through the inlet were equal to the velocity of approach, the static head available would become zero, and also for any other flow condiiton, neglecting friction, the static head available is equal to the head corresponding to the velocity of approach minus the velocity head in the tube. From what has been said, it is now quite obvious that the parallel tube is simply a special case of a diverging tube where the area ratio $\frac{A_1}{A_2}$ is unity at normal capacity.

In Figure 5 is shown a comparison of diverging tubes of various area ratios. In this figure and throughout this paper, what is

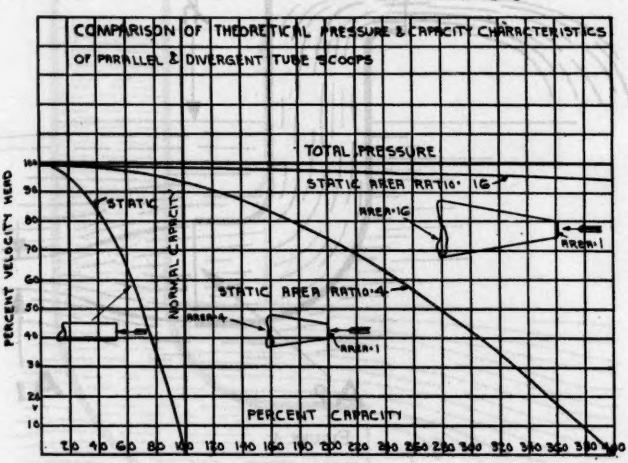


FIGURE 5.

termed the normal capacity of the tube or scoop is the volume corresponding to the inlet area of the tube multiplied by the velocity of approach, that is, the velocity of water past the ship. In the case of the parallel tube, the static head decreases as the quantity passed increases, until, as previously seen from the formula, the static head becomes zero when the velocity through the tube is equal to the velocity of approach, that is, at normal capacity.

In Figure 5 is shown a diverging tube with an area ratio of four, in which the velocity head at the enlarged end is 1/16 that

of the parallel tube for any given inlet velocity. Consequently, the static head available with this tube, neglecting friction, at normal capacity is practically 94 per cent of the velocity of approach, and the static head does not fall to zero until four times the normal volume is reached. In the case of the diverging tube with an area ratio of sixteen, the static head does not become zero until sixteen times the normal volume is passing. If the area ratio were made infinity, neglecting friction and assuming that the tube were immersed in a liquid at infinite pressure, the capacity of any diverging tube would become infinite. Of course, owing to friction, this is impossible. Diverging tubes, however, do pass more than the normal capacity when the back pressure is reduced, as will be seen by the results of tests.

At first thought, it would appear that it is impossible to have more than the normal capacity, that is, that corresponding to the velocity of approach, since apparently, if this were so, it would necessitate an efficiency of over 100 per cent, which is impossible. This is not true, however, as the action which takes place is illustrated in Figure 6, as verified by observing the stream lines around the inlet of diverging scoops.

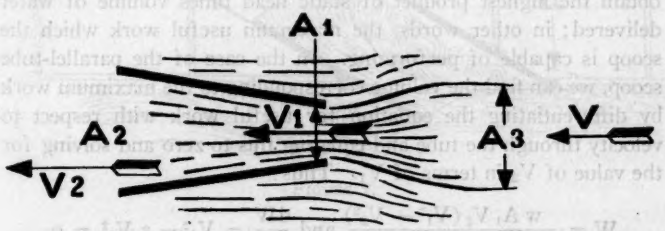


FIGURE 6.

As the diverging tube must necessarily convert the velocity head into static pressure due to the slowing-up of the liquid, if the pressure at the discharge end, or large end, of the tube is reduced below that corresponding to the static pressure resulting from the slowing-up of the liquid, it is obvious that the static pressure at the inlet of the tube must decrease an amount sufficient to maintain this difference of pressure. Consequently, as the pressure at the inlet decreases below that of the surrounding medium, there is a

convergent flow to the inlet, as indicated in Figure 6, and an increase in the velocity through the inlet, with a further increase in static head available, resulting in a further decrease of pressure at the inlet and so forth, until the friction losses establish equilibrium. Under these conditions, the energy available is no longer represented by the tube inlet area, but rather by an area which is larger than the tube inlet, in the ratio of the actual volume to the normal volume, and, consequently, the efficiency under this condition is the actual volume times the static head at the tube discharge divided by the actual volume times the velocity head of approach. This again must necessarily be so because it is observed that the mathematical relation for the static efficiency of the diverging tube depends only upon the ratio of the area. In the case of a straight diverging tube without an elbow at the inlet end, the actual volume passed with zero back pressure at the delivery end of the tube has been found to exceed twice the normal capacity.

When scoops are employed for supplying water to condensers, it is necessary to have sufficient static pressure to overcome the resistance of the condenser tubes, the losses in the water boxes, and pipe friction. Consequently, what we are interested in is to obtain the highest product of static head times volume of water delivered; in other words, the maximum useful work which the scoop is capable of performing. In the case of the parallel-tube scoop, we can find the volume corresponding to the maximum work by differentiating the equation for useful work with respect to velocity through the tube and equating this to zero and solving for the value of V_2 in terms of V_1. Thus:

$$W = \frac{w A_1 V_2 (V_1^2 - V_2^2)}{2g} \text{ and } \frac{dW}{dV_2} = V_1^2 - 3 V_2^2 = 0$$

$$\text{or } V_2 = \frac{V_1}{\sqrt{3}} \text{ and the maximum work becomes}$$

$$W_{max} = \frac{w A_1 V_1^3}{2g \sqrt{3}} \left(1 - \left(\frac{1}{\sqrt{3}}\right)^2\right).$$

In other words, the maximum efficiency which is possible with a parallel scoop, neglecting friction and other losses, is 38.6 per cent and the quantity of water is roughly 58 per cent of the normal.

Since the maximum efficiency of the diverging scoop occurs at normal capacity, it is obvious that the exposed area of a diverging scoop external to the hull need only be half of that of a parallel-tube scoop. As commonly constructed, the parallel scoops which have been employed in the past cannot, of course, approach the theoretical maximum static efficiency, owing to the fact that there is friction, and also the scoops are cut off at an angle, necessitating a turning of the stream in order to enter the scoop.

It would seem quite natural that water passing the ship at velocities of thirty to forty feet per second would not be able to make a sudden twenty or thirty-degree change of direction, with the result that the water would enter the scoop more or less as shown in Figure 7. One would expect, as in Figure 7, that violent

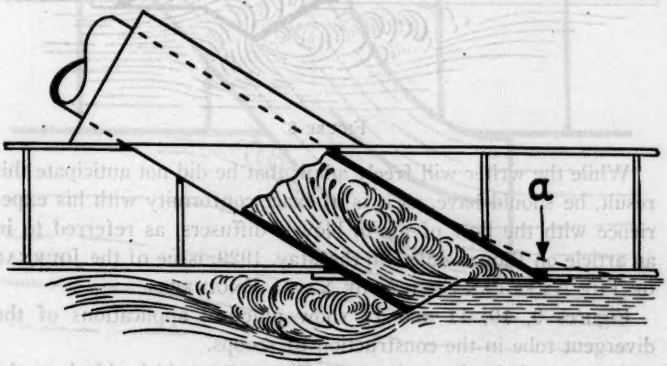


FIGURE 7.

eddies would be set up in the forward entrance area of the scoop and that only the exposed area of the lip would be effective in "picking up" the water. However, while this is what anyone would ordinarily expect, the actual case is quite different, as the experimental model of a standard scoop will disclose.

What actually happens is illustrated in Figure 8, which represents the stream lines as disclosed by experiment. The water enters the forward edge of the scoop, and, contrary to all expectations, the lip, instead of "picking up" the water, actually has water coming out of it. The amount of flow out of and around the lip

of the scoop depends upon the resistance to flow which the scoop is called upon to overcome. However, the surprising fact is that even with the discharge of the scoop open the full bore of the pipe, there is still a small amount of outflow around the edge of the lip.

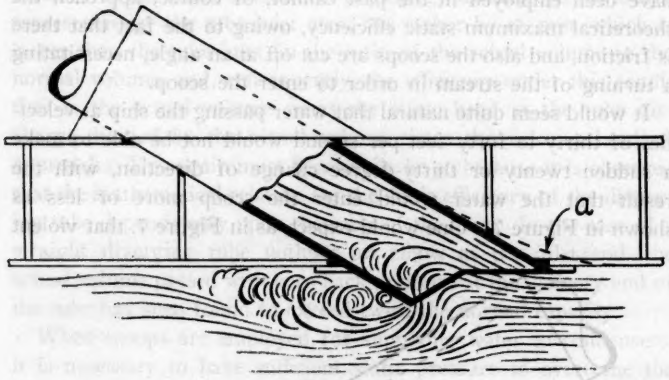


FIGURE 8.

While the writer will freely admit that he did not anticipate this result, he should have, as it is in exact conformity with his experience with the flow of air in blower diffusers, as referred to in an article on Blower Inlets in the May, 1929, issue of the JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS.

Figures 9, 10, 11 and 12 show practical applications of the divergent tube in the construction of scoops.

A scoop of the form shown in Figure 9 would be ideal, as the stream lines enter the scoop in straight lines and the velocity conversion is accomplished before turning an elbow, thus making possible the highest efficiency. This type would be very objectionable, however, because of the rather large structure external to the skin of the ship.

Figure 10 shows a favorable form, the water being turned through only a 45-degree angle before entering the divergent diffuser. Such a design would give an efficiency very close to that of Figure 9. However, it involves the cutting through of at least one frame.

Figures 11 and 12 show designs in which the water is turned through a 90-degree angle before entering the diffuser. An elbow

with " splitters" is employed in that shown in Figure 11, while that in Figure 12 employs the so-called " bladed corner." The " bladed corner" probably is somewhat more efficient than the elbow with " splitters" owing to the fact that there are smaller variations in velocity across the throat of the diffuser, but this form has the objection that the larger number of edges would tend easily to

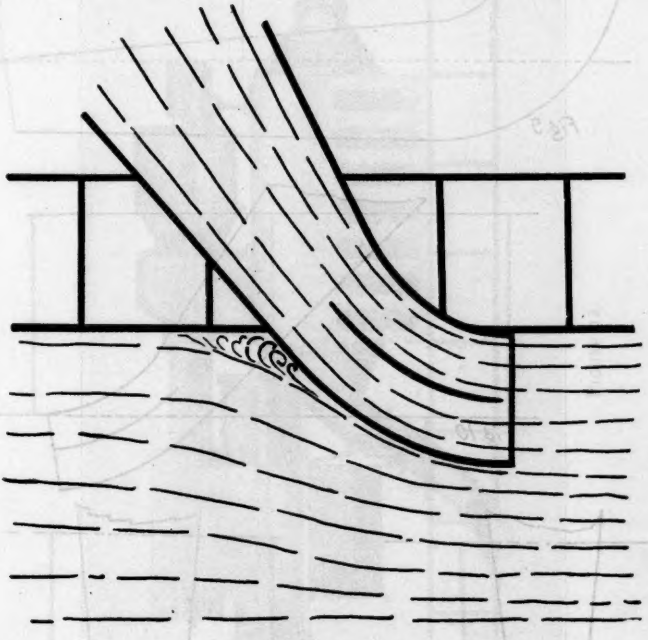


FIGURE 8A.

become fouled with seaweed and other foreign matter. The elbow with two splitters gives nearly as good results and has not the objection of a large number of edges. Neither of these latter designs necessitates the cutting of any frames.

The actual application of the divergent-tube scoop to a condenser is shown in Figure 13, illustrating an underneath condenser fitted with a divergent scoop and a propellor circulating

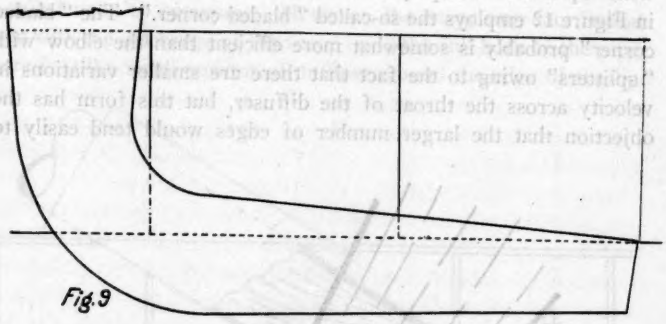


Fig. 9

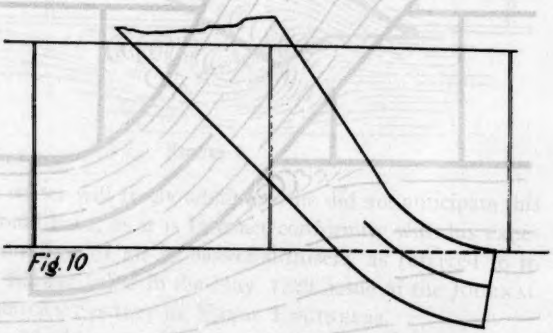


Fig. 10

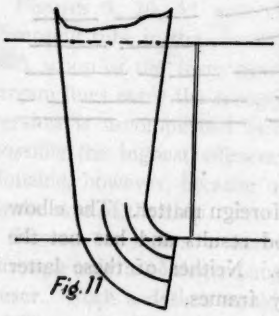


Fig. 11

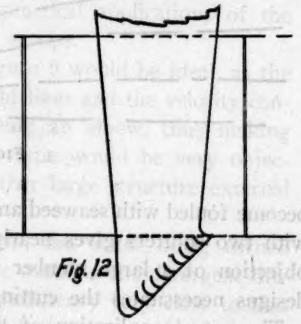


Fig. 12

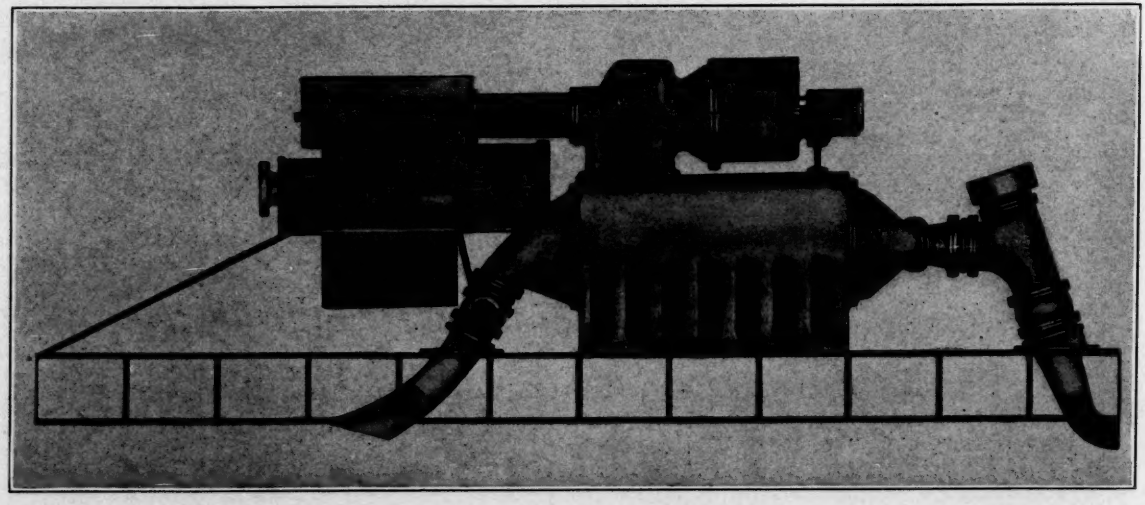


FIGURE 13.

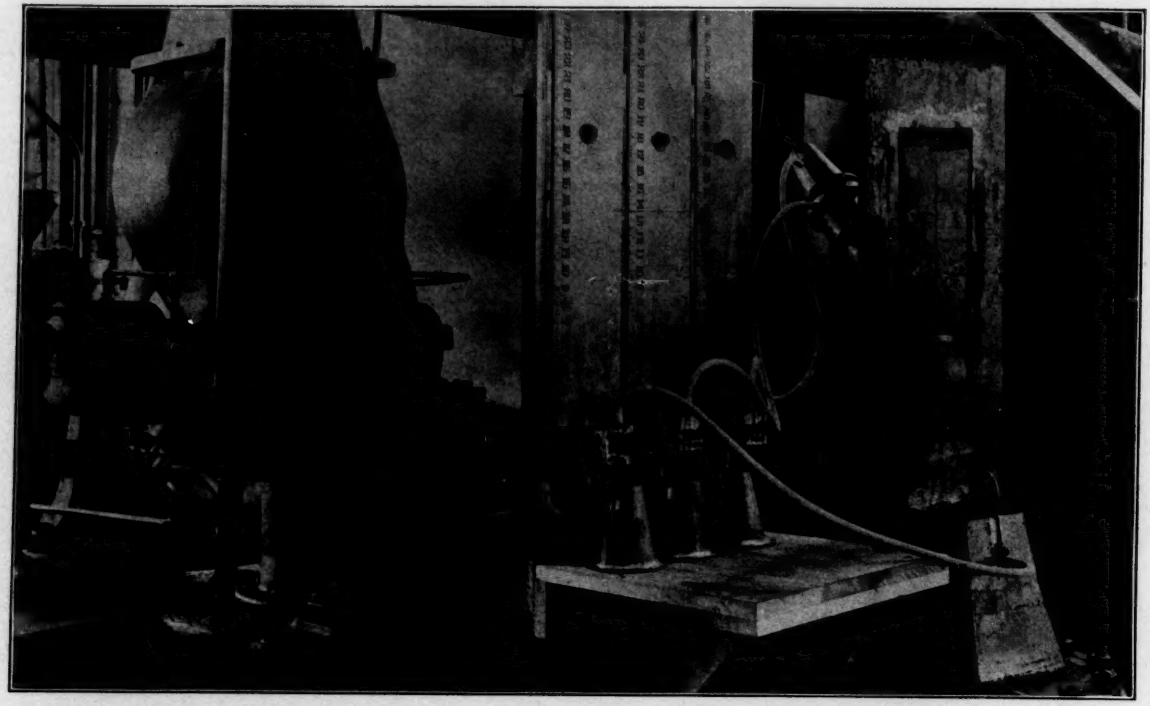


FIGURE 15.

pump for standby. The overall length from scoop inlet to discharge is only a little over half that of the conventional type. Also, because of the higher efficiency, both the injection and discharge pipes are of considerably smaller diameter, thus reducing not only the weight of piping but also the amount of contained water.

EXPERIMENTAL WORK.

As a result of previous experience in the testing of propeller pumps and blowers, the writer had found that it is entirely possible and satisfactory to conduct model tests with air instead of water and that the design constants so derived applied equally well both to pumps and blowers. Consequently, it was decided to employ air for the scoop experiments now to be described.

A centrifugal blower of approximately 6000 cubic feet capacity at 3 pounds pressure was available, and this was used to furnish the necessary air.

One form of divergent-tube scoop tested is shown in Figure 14. Air at various pressures was supplied to a 5-inch square nozzle,

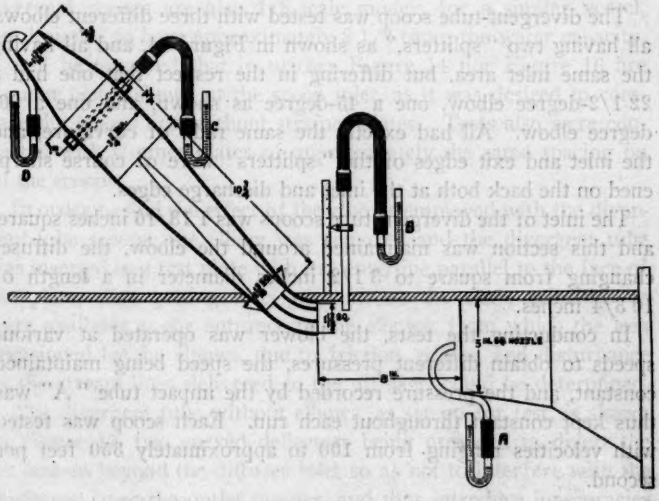


FIGURE 14.

FIGURE 13.

one wall of which was a flat plate extending approximately 30 inches beyond the end of the nozzle, and the various scoops to be tested were inserted in this plate with the inlet located on the centerline of the nozzle and 8 inches from the mouth. A 3/32-inch diameter impact tube was inserted on the centerline of the nozzle and one inch in front of the scoop inlet, connected to a guage "B," and velocity readings were taken at each 1/8 inch outward from the surface of the plate, thus obtaining the velocity at different layers in order to determine the mean velocity head available at the scoop inlet.

Beyond the end of the divergent tube, a parallel portion of pipe was inserted with a piezometer ring connected to the gauge "C," and nozzles similar to that shown, but of various sizes, were attached to the end of the straight section of pipe one inch beyond the piezometer ring. An impact tube having its open end substantially over the centerline of the nozzle and one-half inch in front of the mouth was permanently soldered to each nozzle. An impact tube was also permanently soldered to the 5-inch square test nozzle, this impact tube being located on the centerline one inch in front of the nozzle and 3 1/2 inches from the face of the plate.

The divergent-tube scoop was tested with three different elbows, all having two "splitters," as shown in Figure 14, and all having the same inlet area, but differing in the respect that one had a 22 1/2-degree elbow, one a 45-degree as shown, and one a 90-degree elbow. All had exactly the same radii of curvature, and the inlet and exit edges of the "splitters" were of course sharpened on the back both at the inlet and discharge edges.

The inlet of the divergent-tube scoops was 1 13/16 inches square, and this section was maintained around the elbow, the diffuser changing from square to 3 1/2 inches diameter in a length of 10 3/4 inches.

In conducting the tests, the blower was operated at various speeds to obtain different pressures, the speed being maintained constant, and the pressure recorded by the impact tube "A" was thus kept constant throughout each run. Each scoop was tested with velocities ranging from 100 to approximately 350 feet per second.

All the pressures were read with water gauges and the temperature of the air was taken at the discharge from the 5-inch test nozzle.

A photograph of the test apparatus is shown in Figure 15.

A scoop of the conventional type is shown in Figure 16, this being a 1/8-scale model of a scoop for a 32 1/2-knot vessel. The

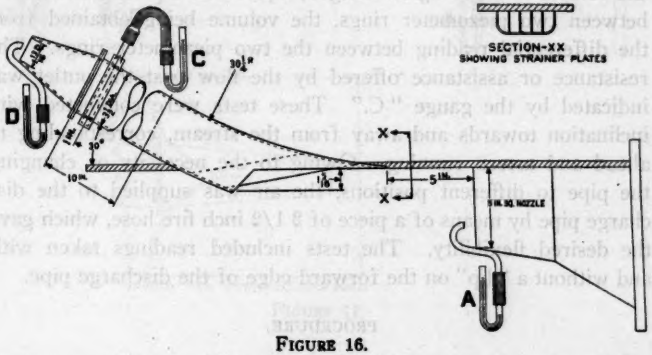


FIGURE 16.

divergent scoops are also 1/8-scale models for a similar vessel, but designed to have approximately 2 1/2 times the water capacity. It will be observed that in neither Figure 14 nor Figure 16 are strainer plates shown at the scoop inlet, as it was desired to compare the scoops first without strainer plates. Tests also were conducted with strainer plates of approximately the same spacing on all the scoops.

In order to find the effect of the elbows employed with the divergent-tube scoops, the elbow was removed and the divergent tube was inserted in a test plate with its centerline parallel to the face of the plate, thus getting tests with air at the same velocities as were available at the entrance to the elbows, from which the loss occasioned by the elbows, due to friction, eddies, and disturbance of the stream lines delivered to the diffuser, could be determined.

The divergent tube without elbows, as set up for test, is shown in Figure 17, two curved deflectors being provided to divert the air stream beyond the diffuser inlet so as not to interfere with the discharge from the outlet nozzles, and thus introduce inaccuracies in the readings of gauges " C" and " D."

It was desired to find the effect of the velocity of the water passing the ship on the flow from the discharge pipe. To obtain this, an arrangement shown in Figure 18 was employed. The discharge pipe was arranged to discharge at 90 degrees, 45 degrees, and 22 1/2 degrees with the face of the test plate, the discharge pipe being 3 1/2 inches diameter and 8 inches long. To obtain the volume of air being discharged, a perforated plate was inserted between two piezometer rings, the volume being obtained from the differential reading between the two piezometer rings. The resistance or assistance offered by the flow past the outlet was indicated by the gauge "C." These tests were conducted with inclination towards and away from the stream, corresponding to ahead and astern running. Owing to the necessity of changing the pipe to different positions, the air was supplied to the discharge pipe by means of a piece of 2 1/2 inch fire hose, which gave the desired flexibility. The tests included readings taken with and without a "lip" on the forward edge of the discharge pipe.

PROCEDURE.

First, a number of readings of gauges "A" and "B" were taken at different velocities to obtain the relation between "A" and "B." These results were plotted as in Figure 19. To obtain the mean head at various distances from the plate, the area under the curves for the different velocities was found by means of a planimeter, and the mean ordinate so obtained is given under each curve in per cent of the stream velocity head. From the mean values of Figure 19 by cross-plotting, Figure 20 was obtained, which gives the mean velocity head corresponding to any stream velocity head —that is, gauge "A."

As the variations of velocity at different distances from the plate are a function of the roughness of the plate only, the relationship between the gauge reading "A" and the mean head available at any given distance from the plate is constant; hence, it was not necessary to take readings of gauge "B" for every test. Checks were taken, however, from time to time to ascertain that no change had taken place.

During the tests, only gauges "A," "C," and "D" were read, and the values of "C" and "D" were plotted as ordinates with

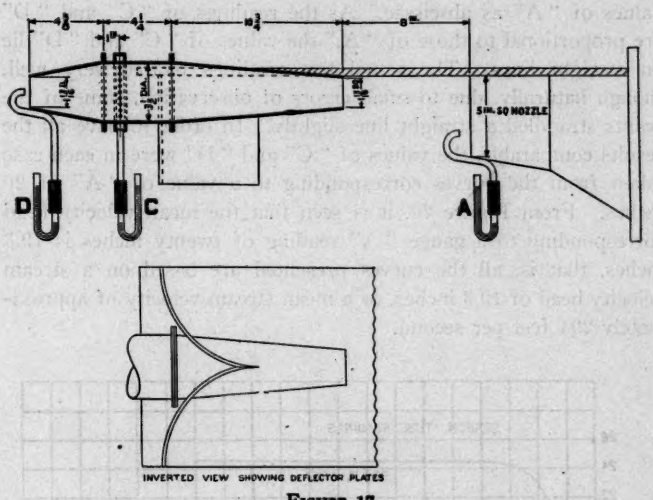


FIGURE 17.

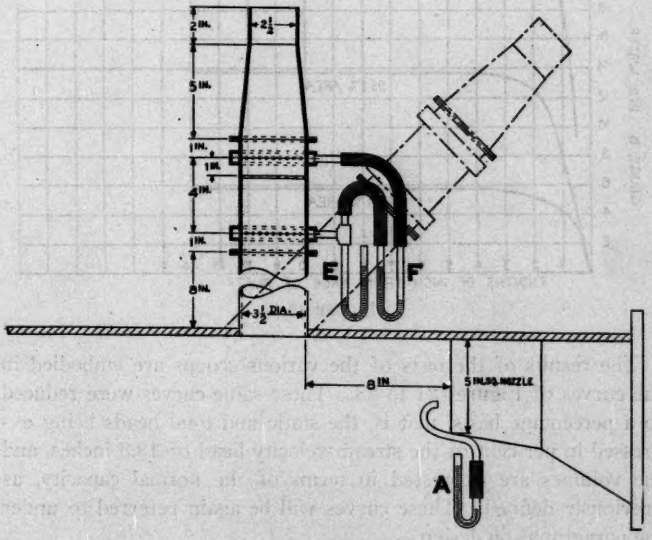


FIGURE 18.

values of "A" as abscissae. As the readings of "C" and "D" are proportional to those of "A," the values of "C" and "D" lie on straight lines. The actual test readings plotted very well, though naturally, due to small errors of observation, some of the points straddled a straight line slightly. In order to have all the results comparable, the values of "C" and "D" were in each case taken from the curves corresponding to a value of "A" of 20 inches. From Figure 20, it is seen that the mean velocity head corresponding to a gauge "A" reading of twenty inches is 19.3 inches, that is, all the curves presented are based on a stream velocity head of 19.3 inches, or a mean stream velocity of approximately 294 feet per second.

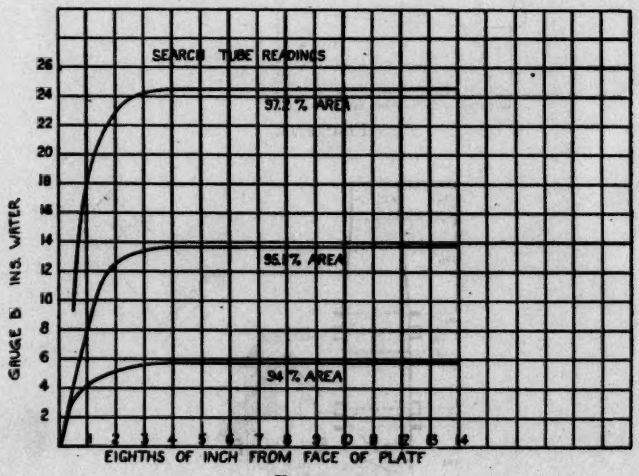


FIGURE 19.

The results of the tests of the various scoops are embodied in the curves of Figures 21 to 28. These same curves were reduced to a percentage basis, that is, the static and total heads being expressed in per cent of the stream velocity head of 19.3 inches, and the volumes are expressed in terms of the normal capacity, as previously defined. These curves will be again referred to under the paragraphs on design.

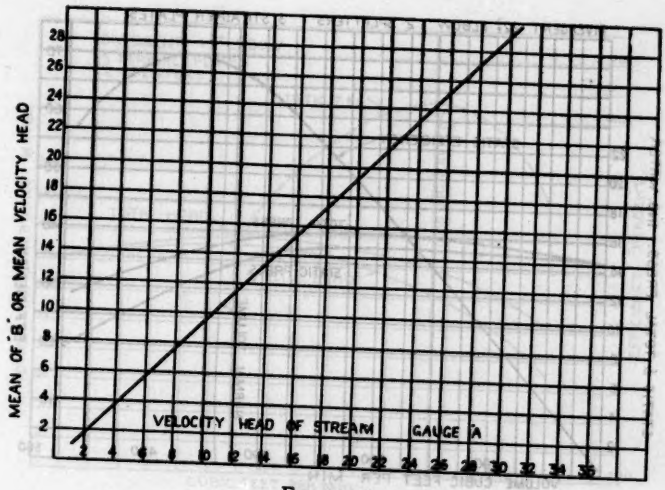


FIGURE 20.

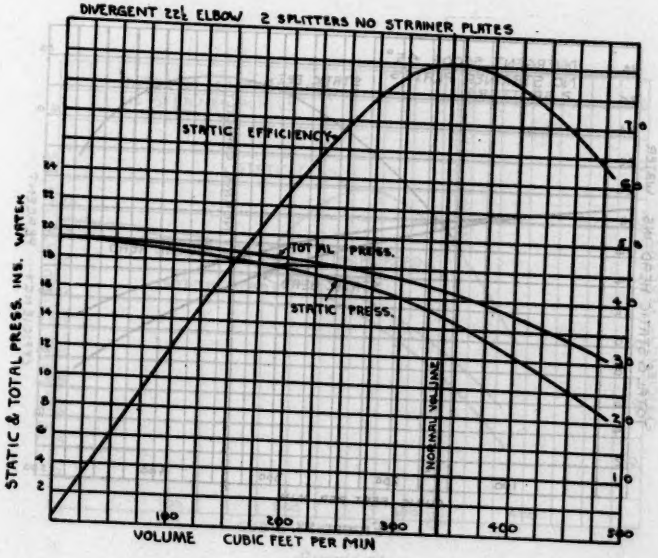


FIGURE 21.

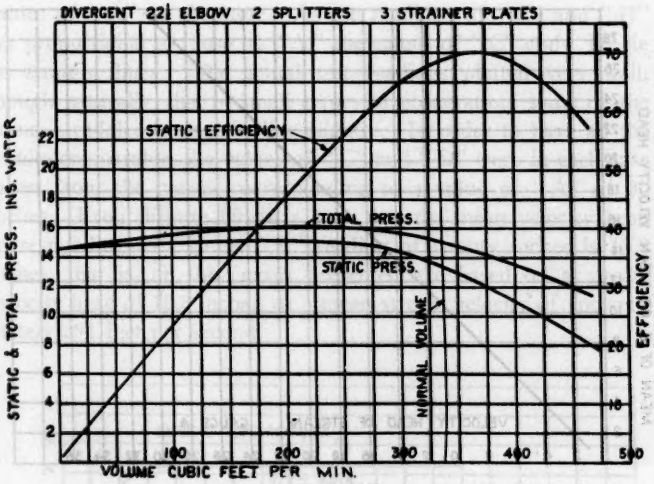


FIGURE 22.

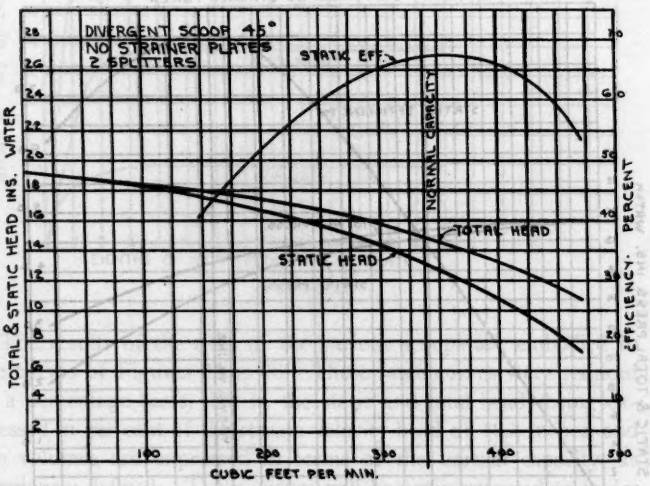


FIGURE 23.

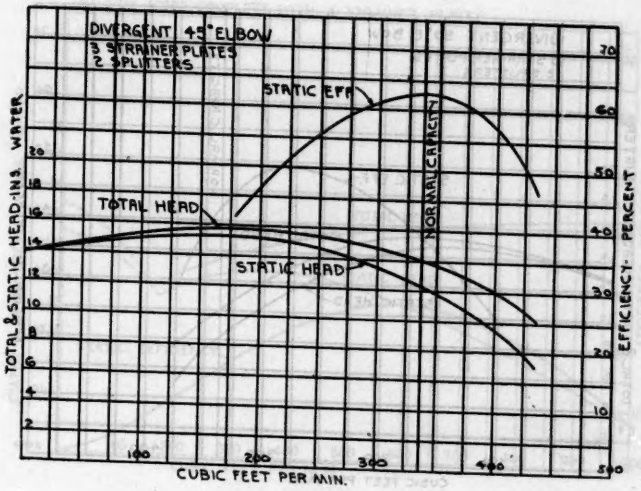


FIGURE 24.

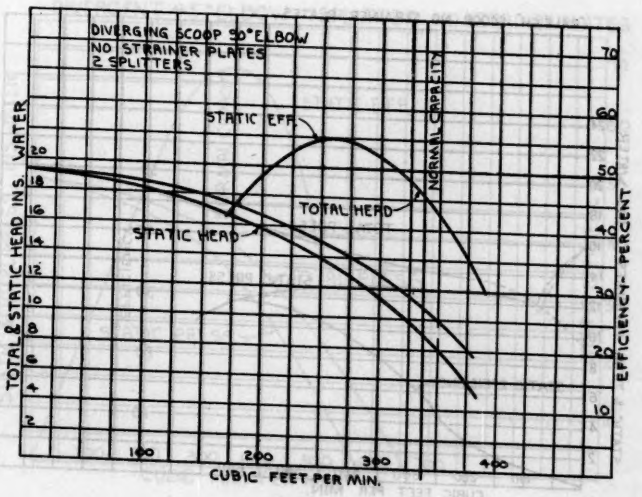


FIGURE 25.

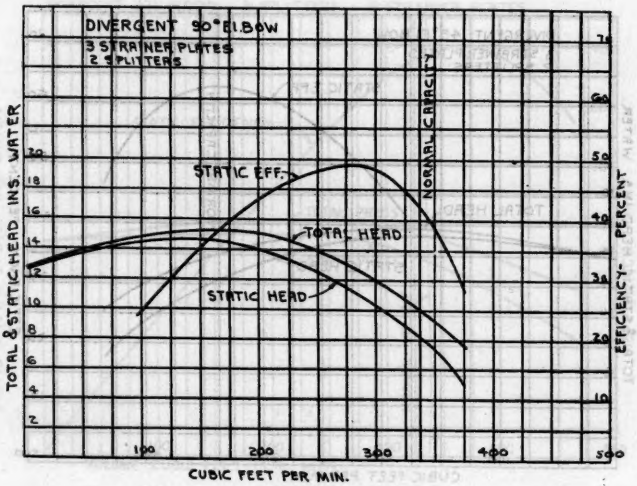


FIGURE 26.

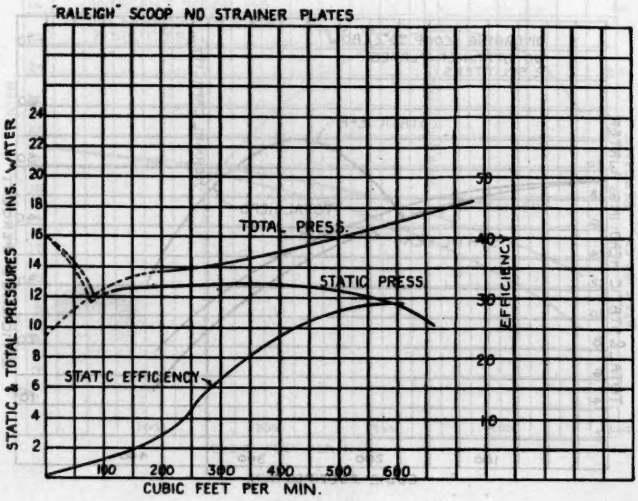


FIGURE 27.

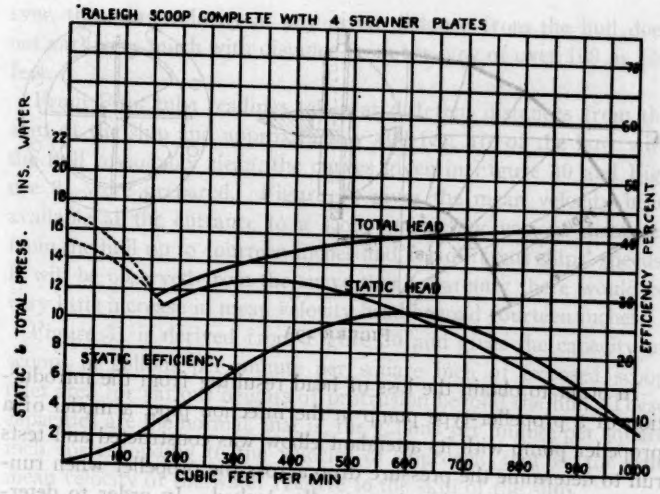


FIGURE 28.

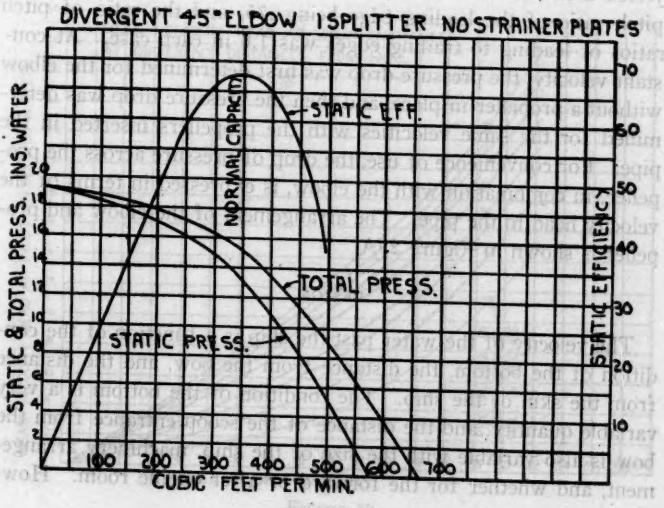


FIGURE 29.

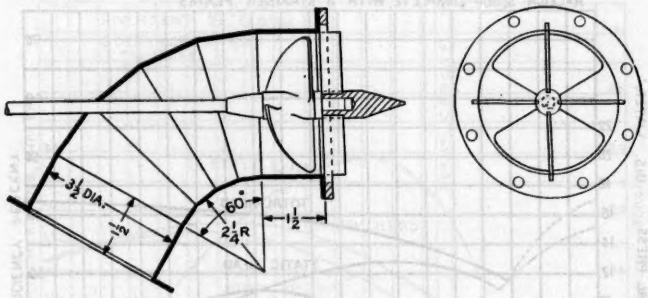


FIGURE 29A.

In order to obtain the loss of head resulting from the introduction of a propeller-type pump in the injection pipe, a model of a propeller pump with its attendant elbow was constructed and tests run to determine the pressure drop across the propeller when running idle, and also with the propeller locked. In order to determine the effect of various projected area ratios, three different propellers of the writer's standard design were tested, having projected area ratios of twenty, forty, and sixty-three per cent, the pitch ratio of the leading edge being .75, and the ratio of pitch ratios of leading to trailing edges was 1.6 in each case. At constant velocity, the pressure drop was first determined for the elbow without a propeller in place, and then the pressure drop was determined for the same velocities with the propellers inserted in the pipe. For convenience of use, the drop of pressure across the propeller, in combination with the elbow, is expressed in terms of the velocity head in the pipe. The arrangement of the elbow and propeller is shown in Figure 29A.

DESIGN.

The velocity of the water past the ship is a function of the condition of the bottom, the distance from the bow, and the distance from the skin of the ship. The condition of the bottom is a very variable quantity, and the distance of the scoop entrance from the bow is also variable with the size of the ship, machinery arrangement, and whether for the forward or after engine room. How-

ever, the mean velocity at any given distance from the hull does not vary very much with distance from the bow of over 100 or 150 feet.

From Pitot tube readings taken at different distances from the skin of the ship and approximately 200 feet aft of the bow with the hull reasonably clean, the curves given in Figure 30 and Figure 31 were prepared. Figure 30 gives the mean velocity head available at the entrance to a scoop projecting various distances from the hull up to fourteen inches and for different ships' speeds. It will be observed from the curve that apparently there would be very little increase in mean velocity head beyond fourteen inches.

Figure 31 is derived from Figure 30 and gives the capacity of scoops in gallons per minute per square inch of exposed scoop inlet area for various heights of protrusion beyond the hull. These capacities are the normal, that is, the gallons per minute per square inch corresponding to the velocity of approach, that is, the true mean velocity of the water relative to the skin of the ship.

The preceding figures, combined with the scoop data presented in Figures 32 to 39, enable the designer to predict accurately the performance of any given type of scoop.

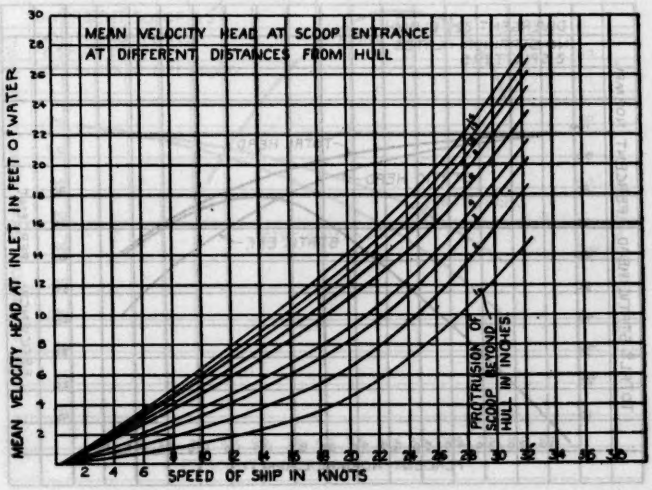

MEAN VELOCITY HEAD AT SCOOP ENTRANCE
AT DIFFERENT DISTANCES FROM HULL
MEAN VELOCITY HEAD AT INLET IN FEET OF WATER
SPEED OF SHIP IN KNOTS
PROTRUSION OF SCOOP BEYOND HULL IN INCHES

FIGURE 30.

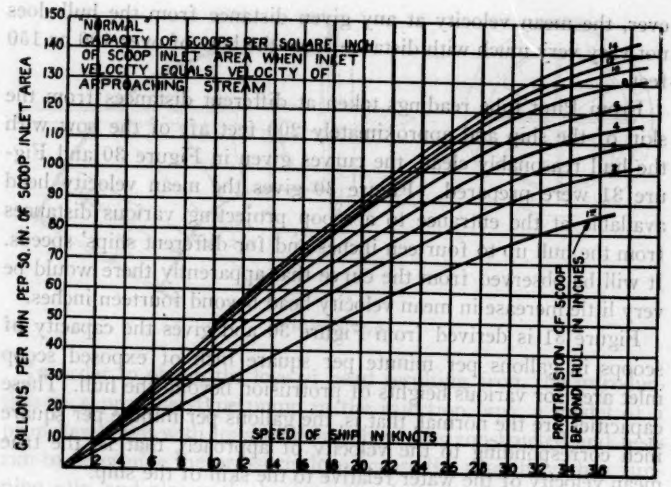

"NORMAL" CAPACITY OF SCOOPS PER SQUARE INCH OF SCOOP INLET AREA WHEN INLET VELOCITY EQUALS VELOCITY OF APPROACHING STREAM
GALLONS PER MIN PER SQ. IN. OF SCOOP INLET AREA
SPEED OF SHIP IN KNOTS
PROTRUSION OF SCOOP BEYOND HULL IN INCHES.

FIGURE 31.

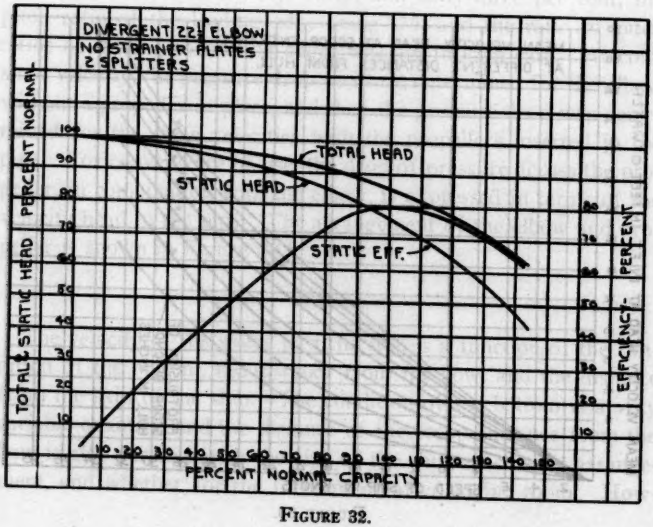


FIGURE 32.

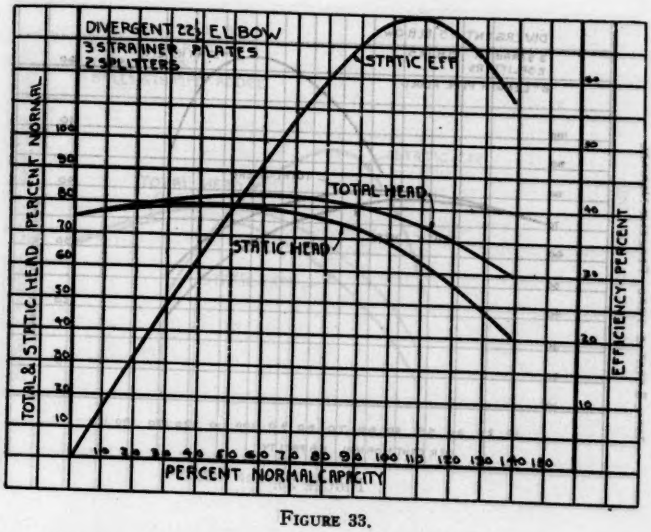


FIGURE 33.

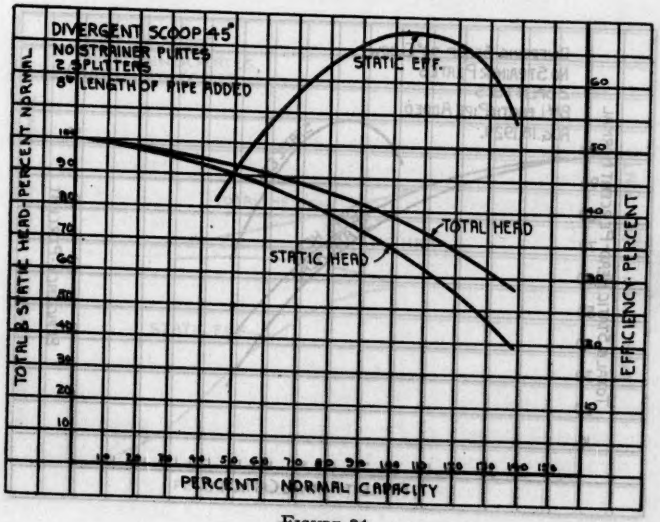


FIGURE 34.

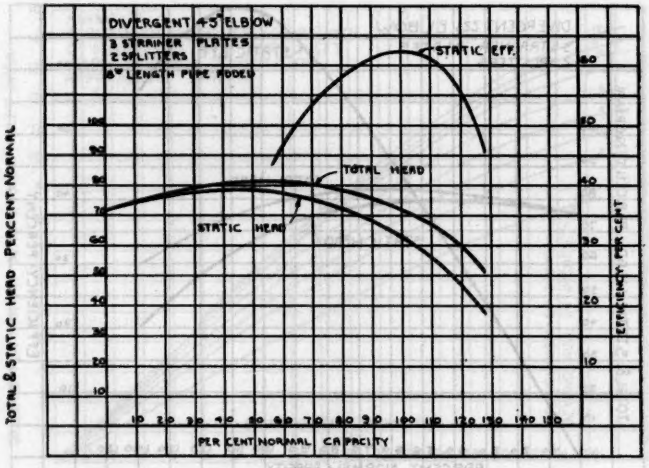


FIGURE 35.

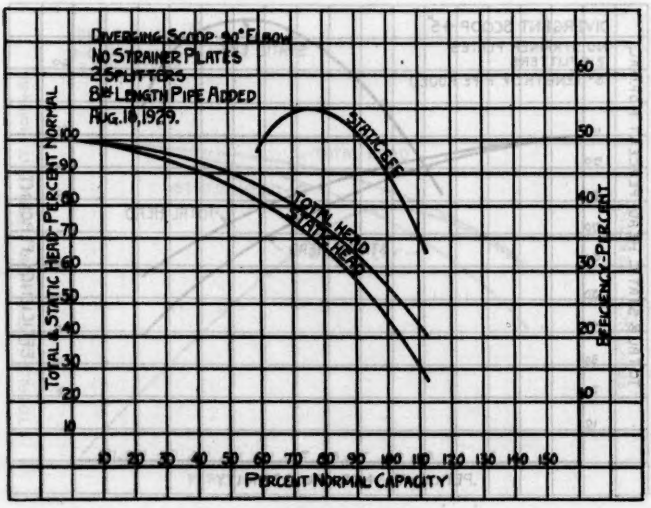


FIGURE 36.

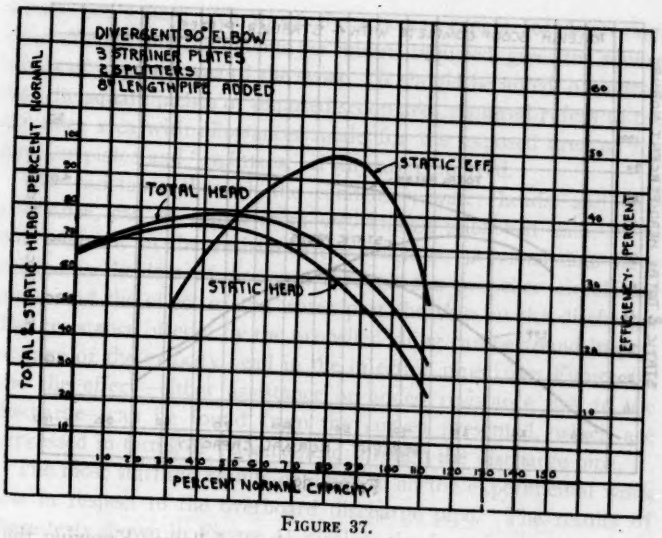


FIGURE 37.

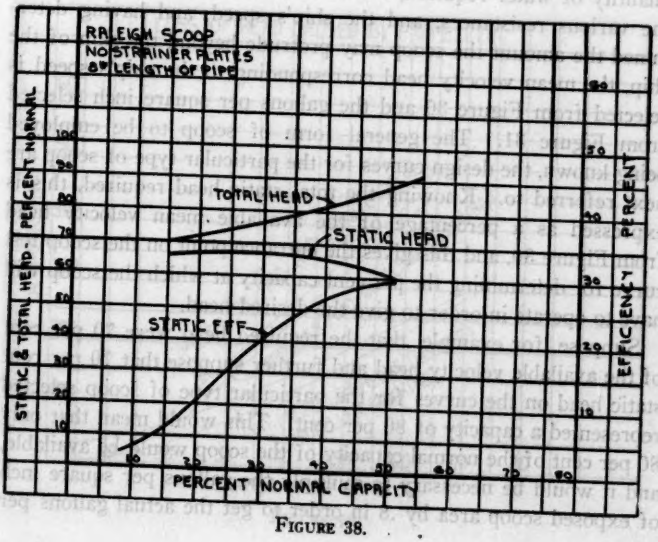


FIGURE 38.

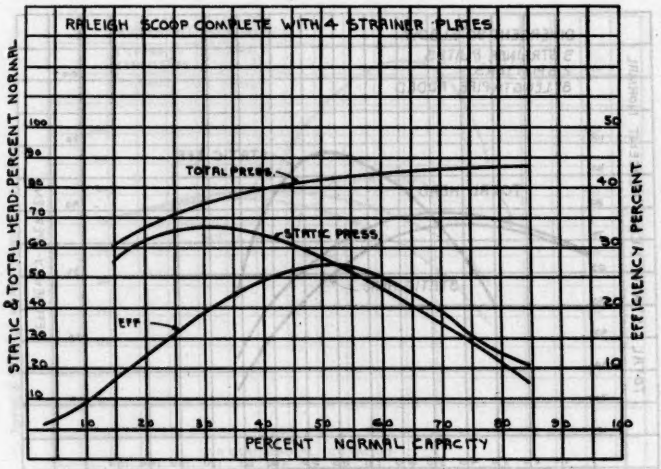


FIGURE 39.

The method of using these curves is as follows: Knowing the quantity of water required, the static head necessary to overcome the various resistances, and the ship's speed, and having determined the amount the scoop may protrude beyond the skin of the ship, the mean velocity head corresponding to the ship's speed is selected from Figure 30 and the gallons per square inch selected from Figure 31. The general form of scoop to be employed being known, the design curves for the particular type of scoop are next referred to. Knowing the total static head required, this is expressed as a percentage of the available mean velocity head from Figure 30, and this gives the entrance point on the scoop test curve for determining the per cent capacity at which the scoop will have to operate in order to give the desired head.

Suppose, for example, that the required head were 70 per cent of the available velocity head and further suppose that 70 per cent static head on the curves for the particular type of scoop selected represented a capacity of 80 per cent. This would mean that only 80 per cent of the normal capacity of the scoop would be available, and it would be necessary to multiply the gallons per square inch of exposed scoop area by .8 in order to get the actual gallons per

minute per square inch which the scoop is capable of delivering. This, divided into the gallons per minute required, gives the square inches of exposed area of the scoop. It should be noted, of course, that the square inches of exposed scoop area required refers to the net inlet area with allowances made for the exposed area of the strainer plates and " splitters," if any are employed.

The resistance through the condenser, water heads, and pipe resistance, can be found from well-known published data. The only element in the resistance on which no previous data have existed is the loss of pressure through the propeller circulating pump and the effect of the water past the ship on the discharge. The resistance offered by the propeller idling can be found in percentage of the velocity head in the injection pipe from Figure 40, and the effect—either assistance or added resistance due to the discharge—can be found from the curves presented, which are expressed in terms of velocity head through the discharge pipe.

The most startling results obtained in all the experimental work was in respect to the overboard discharge pipe. The results of these tests shown in Figure 41 disclose the fact, for instance, that

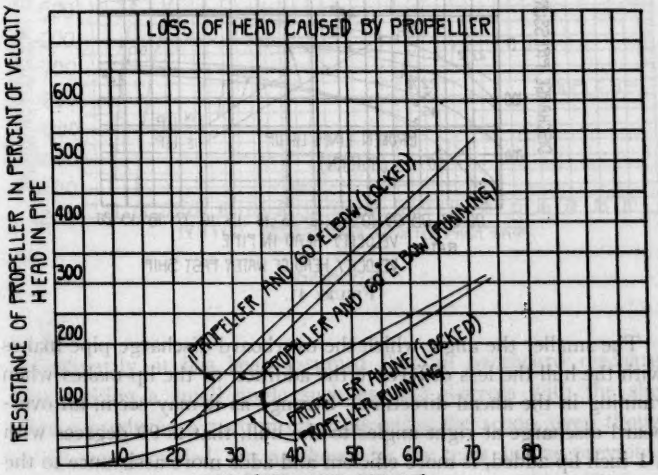


FIGURE 40.

when going in the ahead direction an overboard discharge pipe without a lip imposes a back pressure retarding the flow of water, regardless of the angle of inclination, and that while the smaller angle of discharge gives less back pressure when running in the ahead direction it gives a correspondingly higher back pressure when running astern.

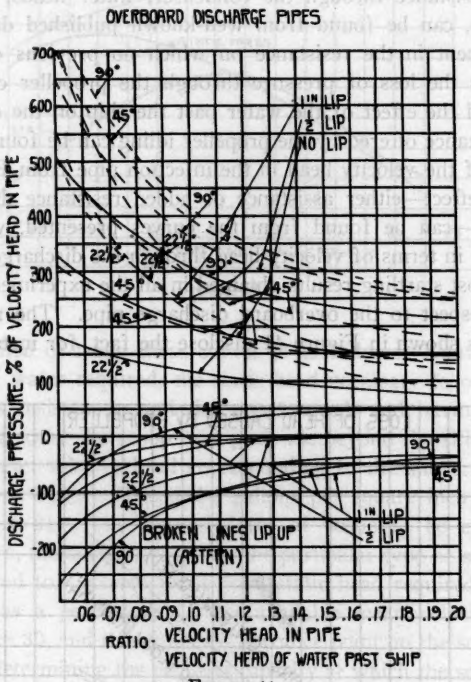


FIGURE 41.

The smaller the angle which the overboard discharge pipe makes with the hull the less difference the addition of the lip makes when running in the ahead direction. Strange as it may seem, an overboard discharge at right angles to the hull, that is 90 degrees with a 1-inch lip added, is more efficient and adds more assistance to the flow than the discharges making a finer angle with the skin of the ship. In Figure 41 it should be borne in mind that the 1/2-inch lip

and 1-inch lip referred to are on a 3 1/2-inch discharge pipe, that is the protrusion measured at right angles to the hull. In order to use this data for general design the height of the lip should be taken in proportion to the diameter, that is, the 1/2-inch lip corresponds to a lip-protruding 14.3 per cent of the diameter of the discharge pipe beyond the hull and the 1-inch lip 28.6 per cent of the diameter. Used in this manner, it is applicable to discharge pipes of any diameter.

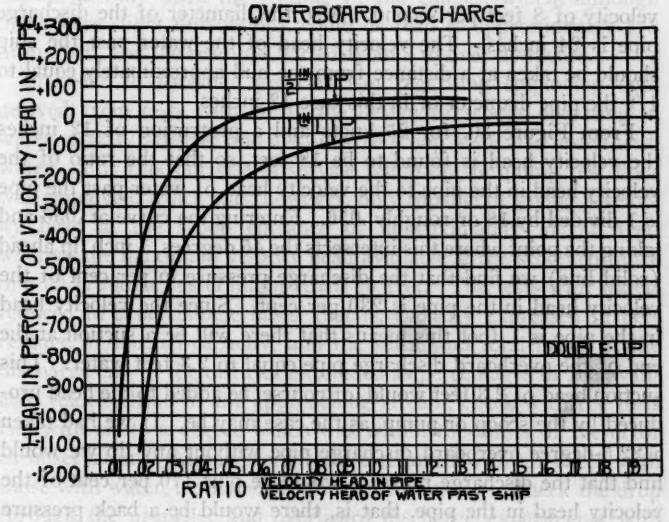


FIGURE 42.

Ordinarily, it has been assumed that an overboard discharge pipe making a small angle such as 22.5 degrees with the shell plating produced a suction effect assisting the scoop, even though no lip was provided, but as shown by the tests and as previously mentioned instead of a suction there is a back pressure. The tests show that the 90 degrees overboard discharge (which is the short-

est and lightest) when fitted with a 28 per cent diameter lip produces a greater suction at all ship speeds than a 22.5 degrees.

The method of plotting the results of the overboard discharge tests is rather unusual and may be slightly confusing, but was necessitated by the fact that it was the only system of plotting which the writer could devise which rendered all the results independent of the units and medium employed. It is, however, very convenient for practical use. For example, supposing that a ship has a speed of 25 knots equipped with 45 degrees overboard discharge with a 14.3 per cent diameter lip and the velocity head selected in the overboard discharge is 1 foot corresponding to a velocity of 8 feet per second. Also the diameter of the discharge pipe is 24 inches. The velocity head of the water past the ship should be taken at a distance from the hull approximately equal to 1/2 the pipe diameter or in this case 12 inches.

From Figure 30, for 25 knots and a protrusion of 12 inches the velocity head is found to be 18 feet, so that the ratio of the velocity head in the pipe to the velocity head of water past the pipe is 1 divided by 18 or roughly .056. Entering the curve at .056 and taking the point where this intersects the 45 degrees, 1 inch lip ahead (solid line) we find that the discharge pressure in per cent of the velocity head in the pipe is 230 per cent. Since the velocity head in the pipe is 1 foot this means that there will be a suction at the end of the overboard discharge pipe equal to 2.3 feet water. This suction head of 2.3 feet would, of course, be added to the head produced by the scoop or pump, as the case may be. If we had taken a 22.5-degree overboard discharge pipe without any lip we would find that the discharge pressure would be plus 170 per cent of the velocity head in the pipe, that is, there would be a back pressure impeding the flow equal to 1.7 feet of water and this would have to be subtracted from the head produced by the scoop or pump.

W. B. Flanders has suggested the desirability of obtaining an overboard discharge which would have the same characteristics in either direction of motion, which can be accomplished by using a 90-degree discharge with two lips. This was tested and the results are incorporated in Figure 42. In this case, the forward and after lips taper down to nothing on the athwartship diameter. The same results could be obtained as far as the pressure characteristics of

the discharge are concerned by simply letting the pipe extend beyond the ship an amount equal to the lip but this would add considerably to the resistance offered by the discharge pipe since in the double lip arrangement the stream issuing splits and the greater part is discharged through the two notches, flattening out against the skin.

CONCLUSION.

It is safe to say that the probable accuracy of the tests submitted is of the order of plus or minus one per cent. The accuracy was dependent only upon the reading of the gauges, which, although there were slight fluctuations, could be read to approximately 1/8 of an inch, and the readings varied from 8 to over 20 inches.

In order to forestall any criticism regarding employing data derived from tests with air in the design of scoops operating in water, a 22 1/2-degree and a 45-degree scoop were also tested with water. For this purpose, a service pump on the pump test floor was employed, which was, however, only capable of giving a velocity head at the scoop entrance of approximately 17 1/2 feet, whereas the maximum velocity head available at the scoop entrance in a 32 1/2-knot vessel might reach 30 feet. The tests with water, reduced to a percentage basis, that is, capacity expressed in percentage of normal capacity, and the static and total heads at the discharge end of the diffuser expressed as percentages of the available velocity head of the approaching stream, check with the tests run with air to a degree which was within the limits of the error of observation. As the scoop tests conducted with air check with those with water, it was not considered necessary to check the drop through the propeller and overboard discharge experiments with water.

That the tests conducted with water should check with those run with air simply verifies the mathematical relationships which apply to scoops or divergent tubes, since it will be observed in the various formulas presented in the theoretical discussion that all the expressions for head contain no factors which are dependent upon the medium employed, as long as the head is expressed in feet of the same fluid.

Strictly speaking, from a scientific standpoint, there are factors which should be taken into consideration, such as the scale effect and the variation in kinematic viscosity between water and air. As far as the writer has been able to observe from his experience with pumps and blowers, these do not influence the results to a practical degree within the limits of size which would ever be required.

A number of propeller type pumps up to 44,000-gallons-per-minute capacity have been designed using the constants which were derived from 12-inch propellers operating in air, and these have performed exactly in accordance with the predictions.

Examining the various test curves for different types of scoops and observing the percentage of the velocity head of approach which is available to overcome condenser and other resistances, it seems quite evident that the scoop and condenser must be designed as cooperating parts of a complete system. For any given speed of ship, there is a certain maximum head available for overcoming the condenser resistance, which, if exceeded, will seriously reduce the amount of water passed through the condenser. This has undoubtedly been the cause of the unsatisfactory performance of certain scoop-condenser installations.

The condenser, piping, etc., should be designed to have the minimum of resistance for the quantity of water required, since, for any given speed of ship, the smaller the total resistance, the less exposed area of scoop is required.

While the usual design of the scoop and discharge piping is very much spread out owing to the small angle of inclination with the hull, it is obvious from the theory that it is possible to design a parallel-tube scoop with a right-angle turn coming straight up between frames, the same as is possible with the divergent scoop. However, there is this important difference, that in order to get the same capacity and static pressure, the exposed area of such a scoop with an abrupt turn would have to be from two to three or four times that of a divergent scoop.

The appendage resistance due to the scoops is of course dependent upon the exposed area and the quantity of water delivered. If we examine the stream lines of the usual design of scoop shown in Figure 8, it is obvious that the resistance offered in dragging this through the water very materially exceeds that of a flat plate

of similar area, since there is actually a reverse flow and part of the water is discharged in a forward direction, as well as being deflected around it. For the same exposed area, the divergent scoop operating at normal capacity offers only a very small fraction of the resistance owing to the fact that the water enters the scoop with the full velocity of approach, and consequently, the stream lines around the scoop are entirely undisturbed, as no water is deflected around the inlet, whereas very violent eddies occur behind a scoop of the usual form. Observations of the stream lines around the different types of scoops operating at various capacities demonstrate this in a very striking manner.

Let us compare the efficiency of a scoop as a means of supplying the necessary water for condensation with that of a centrifugal pump. If we assume a centrifugal pump of 80 per cent efficiency driven by an auxiliary turbine having a water rate of 30 pounds per brake horsepower, and a divergent scoop with a 22 1/2-degree elbow having an efficiency of 78 per cent, and that the propulsive efficiency of the vessel is 60 per cent and the water rate per shaft horsepower is 9 pounds, this would be equivalent to a steam consumption of 19 pounds per water horsepower for the scoop against 37 pounds per water horsepower for the centrifugal pump. It will be observed that in both instances the resistance caused by bringing the pumped water to rest relative to the ship has not been included in either case, since this would be the same for both for equal volumes. We may conclude, therefore, that a well-designed divergent scoop is a more efficient means of supplying condensing water than a centrifugal pump driven by an auxiliary turbine, but that a centrifugal pump driven by an electric motor, taking its current from the main generator of an electric-drive ship, would be slightly more efficient than the scoop. The scoop has a great advantage from a reliability standpoint, however, as it eliminates moving parts when under way.

Owing to the fact that it is possible with the divergent scoop to make a very sharp turn into the ship, and because of the higher efficiency, much shorter and smaller diameter injection and discharge pipes may be employed than with scoops of the usual design. This results in a very considerable saving in weight both

of non-ferrous materials and of the water contained in the piping. In the case of a high-speed vessel of about 100,000 horsepower, the saving is something over 50 tons, of which 20 tons is material.

From the standpoint of the hull designer, the most desirable feature is the reduction or entire elimination of frames penetrated by the injection and discharge piping and a possible shortening of the engine room spaces.

FROM GENERAL BOARD TO TRIAL BOARD.
THE EVOLUTION OF A MAN-OF-WAR.

By COMMANDER A. M. CHARLTON, U. S. NAVY, MEMBER.

After my morning inspection of the engineer department, I had climbed up to the navigating bridge to cool off and to collect any gossip that might have come in from other ships. The Battle Fleet was slowly steaming along in the California sunshine, bound north for the summer in Puget Sound. A slim gray ghost came slipping over the horizon, rapidly overhauled us and went by as if we were standing still, an exchanging flutter of flags asking and giving permission to proceed on duty assigned.

"There goes the *Omaha,*" said the Officer-of-the-Deck. "One of those new scouts of ours. She is just finishing up her full power trials before joining the Fleet."

She made a beautiful picture sliding through the water at something more than thirty knots, scarcely disturbing the sea as she passed, the faintest haze of smoke showing from her smoke pipes.

"Yes," continued the Officer-of-the-Deck, "she's turning out nearly a hundred thousand horsepower, and look at her—if it weren't for the stern wave she's pulling along you would hardly know she was moving. Some contrast to the old coal burning battleships which pushed the whole ocean along in front of them and belched out smoke enough to cover a city. These designers are getting better all the time. There is an awful lot I don't savvy about this shipbuilding. There's the *Omaha,* not as long as we are and displacing a quarter as much and yet to get ten knots more than

NOTE:—The author makes no claim for originality in the subject matter of this article. In compiling it the following sources have been consulted: Taylor's "Speed and Power of Ships," Dyson's "Screw Propellers for Hydraulic and Aerial Propulsion," Carmichael's "Practical Ship Production," and lectures before the Post Graduate School by Rear Admiral Schofield, Commander Howard (CC), and Commander Delano.

we do she has to develop three times as much power. How do they decide on their lines, their length, beam and draft? They tell me that it is possible to dope out the speed, power and revolutions of the propeller almost exactly before the ship is built. I suppose it is all in the book somewhere."

I could not enlighten him much at the time but later on I had an opportunity to find out something of what was "in the book" on this subject. When I was ordered to duty in Washington one of my first assignments was to lecture to the post graduates at Annapolis on the mouth-filling subject "Technical Realization by the Bureau of Engineering of the Military Characteristics of New Construction as Laid Down by the General Board." Investigation of the meaning of this formidable title, and its connection with the work of other technical bureaus of the Navy Department disclosed a process of evolution most interesting, and I believe, little known to the service at large. A modern man-of-war has been said to embody more of the accumulation of human knowledge than any other concept of mortal mind. How is this knowledge applied? What goes on from the time the General Board begins to think of a new ship, to the day in which the Trial Board pronounces her built in accordance with the specifications and meeting the guarantees of the contract?

THE GENERAL BOARD.

Long before the design of any particular type of vessel is taken in hand by the various technical Bureaus of the Navy Department, its necessity has been determined by the deliberations of the General Board.

The General Board was established by order of the Secretary of the Navy, John D. Long, on March 13, 1900. The purpose of the Board as stated in the order establishing it was "to insure efficient preparation of the fleet in case of war and for the naval defense of the coast." It is composed of the Chief of Naval Operations, the Major General Commandant of the Marine Corps, the Director of Naval Intelligence, the President of the Naval War College, a Secretary, and five other officers constituting the "Executive Committee" of the Board. The membership of this committee is largely confined to flag officers, who have had command

afloat. The majority of ex-Commanders-in-Chief come to the General Board.

The decision of the Board (in regard to new construction) is arrived at from a consideration of the functions of our Navy in supporting American policies and protecting American interests, both in peace and in war, and from a consideration also of possible opposing factors in the performance of these functions. In arriving at its decision, the Board takes cognizance of the well-known policies of the United States, of crystallized service opinion, and of information obtained from all over the world. In performing this duty, the Board carries out that provision of the Navy Regulations which states that "it (the General Board) shall consider the number and types of ships proper to constitute the fleet, and shall advise the Secretary of the Navy respecting the same, including such increase as may be requisite to be submitted annually to Congress."

The Regulations also state that "when designs are to be prepared for a new ship, the General Board shall submit to the Secretary of the Navy a recommendation as to the military characteristics embodied therein."

Out of many years experience, the General Board, with the approval of the Navy Department, has laid down the following general principles of design applicable to the Navy as a whole:—

To develop the Navy for operations in any part of either ocean;

To make the strength of the Navy for battle of primary importance;

To make the strength of the Navy for oceanwide economic pressure next in importance;

To make superiority of armament in its class an end in view in every type of fighting ship;

To provide for great radius of action in all types of fighting ships.

Rear Admiral Schofield has said "these simple principles were arrived at after long and exhaustive discussions and hearings. They are a part of the approved Naval Policy of the United States. They are the result of an integration of the factors flowing out of our geographic position, the area of greatest naval interest, and the

kind of resistance we might expect in war, or the kind of pressure we might deem it likely will most effectively support our national interests."

The foregoing principles of design may be paraphrased by saying that the United States must have ships which will go farther and faster, stay longer, and outfight any other ships of their class.

The Regulations further state ". . . Such military characteristics shall include a general statement as to the type of ship, speed, radius of action, and armor and armament."

As sent out to the technical Bureaus of the Navy Department, after approval by the Secretary of the Navy, the military characteristics of new construction usually include the following:—

1. Displacement.
2. Armament, including the number and caliber of guns, and methods of mounting them; torpedo tubes, aircraft, etc.
3. Protection.
4. Speed.
5. Endurance, that is cruising radius at a specified speed.
6. Provisions and stores to be carried, for the period of time during which the vessel must be self-sustaining.
7. Communication equipment—radio, signalling devices, etc.
8. Miscellaneous characteristics, depending on the type of vessel, which are considered necessary to emphasize.

PRELIMINARY STUDY.

Before the final determination of these characteristics is made by the General Board, a great deal of preliminary calculation and investigation has been undertaken, principally by the Bureau of Construction and Repair which is charged with the preparation of preliminary plans. During the first hearings of the General Board in regard to a new vessel, representatives of the Bureau of Construction and Repair (and frequently those of Engineering and Ordnance) are present to answer technical questions, and from present knowledge and past experience to advise the Board whether certain conditions are possible or impossible of fulfillment. For example, the Treaty for the Limitation of Armaments sets a maximum of 10,000 tons displacement for auxiliary cruisers and

limited their guns to 8-inch caliber. It would be highly desirable if such a vessel could be built to make 40 knots, carry 12 8-inch guns and have 6 inches of armor for protection. With our present knowledge of metallurgy, a super-vessel with these characteristics is quite impossible on a displacement of 10,000 tons. So there must be a compromise. The various alternatives possible are stated by the Bureau of Construction and Repair, but the General Board makes the final decision as to the compromise to be submitted to the Secretary of the Navy for approval.

How does one know that a cruiser with the desirable characteristics mentioned above is impossible and that certain other characteristics are possible?

BASES OF CALCULATION.

Knowing that the friction of the water against the skin of the vessel and the disturbance set up by the ship as it passes through the water (and air) demand power to overcome them, the immediate concern is an underwater form which will necessitate the least power for a given speed.

Although experiments on the resistances of surfaces in water were made in the eighteenth century, the first work of importance is considered that of Mr. William Froude, in 1871.

He had H.M.S. *Greyhound* towed by another vessel at various speeds and displacements and measured the resistances on the tow-line. He found that at higher speeds the resistance increased much more rapidly than at the lower, varying with the square of the speed at low speeds to the fourth power of the speed at high speeds. As the power necessary to tow the vessel is equal to the resistance times the velocity, we have power varying with the cube of the speed at low speeds and with the fifth power at high speeds. Thus a ship at 40 knots may require 4 times the power that she does at 30 knots. Froude also found that the power on the tow-rope at a given speed varied considerably from the power developed by the engines of the vessel at the same speed. In other words the efficiency of the propeller has to be considered. The power necessary to tow a vessel is called the effective horsepower (E.H.P.) while that necessary to put into the propeller shafts which turn

the propellers and hence drive the ship ahead is the shaft horsepower (S.H.P.). The ratio $\frac{\text{E. H. P.}}{\text{S. H. P.}}$ is the propulsive coefficient of the propeller.

The resistance encountered in driving a ship through the water is made up of several factors:—

Skin Resistance. Water not being frictionless, its motion past a ship involves a certain amount of frictional drag, the resistance of the surface involving an equal and opposite pull upon the water. It is nearly always the most important factor of the total resistance. The determination of the skin resistance of ships is based entirely upon experiments made with thin, comparatively small planes moving endwise through the water. The classical experiments in this connection were made by Froude who towed boards of various lengths coated with different materials at several speeds through the water of a model tank. The measurements of the resistance showed that it varied with the coating of the board, the amount of wetted surface, and the speed with which it was towed, or $R_f = fSV^{1.83}$ where:

R_f=Frictional resistance
f=Coefficient of friction
S=Wetted surface
V=Speed in knots.

For a vessel of a given displacement and a certain frictional quality the only factor affecting the frictional resistance at any one speed is the length. The greater the length for a given displacement the greater the frictional resistance, because it varies with the wetted surface and this varies with the length.

Eddy Resistance. The motion through the water of a blunt or square stern post or of a short and thick strut arm, etc., is accompanied by much resistance. Such resistance is termed "Eddy resistance." With proper design it is in most cases a very minor factor in the total resistance.

Air Resistance is that offered by the part of the vessel above the surface of the water. It also is of minor importance.

Wave Resistance. The motion of a vessel through the water sets up a series of surface waves. The energy for their production and propagation is of course communicated from the ship. At low

speeds the wave resistance is a very small part of the total resistance, but at speeds which are very high for the vessel's length, the wave resistance may be as much as 60 per cent of the total. It is seldom advisable to adopt a design where the wave resistance is as much as 50 per cent of the total. The wave making resistance is practically impossible to evaluate mathematically but it can be arrived at from the use of models. Modern ideas of the resistance of ships are based largely upon the Law of Comparison or Froude's Law, as it is generally called, which connects the resistance of similar vessels. The use of this law permits the determination, with fair accuracy, of the resistance of a full sized ship from the experimentally determined resistance of a small model of the same ship. Froude's Law may be stated "If two ships are exactly similar and 'N' is the ratio of their corresponding linear dimensions, then if they be run at speeds proportional to the $\sqrt{N}$ the ratio of the corresponding resistances will be N^3". For example, for a ship 500 feet long with a speed of 16 knots, suppose we take a model 20 feet long, N in this case is $\frac{500}{20}$ or 25. Then at a speed of 3.2 knots $\left(\frac{16}{\sqrt{25}}\right)$ of the model we will get a certain resistance "R." The resistance of the full sized ship at a speed of 16 knots will be 25^3R or 15625 R. This Law of comparison does not, however, apply to the skin resistance, but only to the residual resistance, *i.e.*, eddy, air and wave resistance. It is necessary, therefore, in working from a model, to measure the total resistance, subtract the skin resistance, and then step up the residual resistance to that of the full sized vessel. To this resistance is added the skin resistance of the full sized ship, the sum of the two being the total resistance.

MODEL BASIN.

Inasmuch as model results are the basis of calculation, the Model Basin at the Washington Navy Yard is a very important adjunct of the Bureau of Construction and Repair. It is some 470 feet long by 42 feet wide, and 14 feet deep, and contains about two million gallons of water. Models used are generally of California redwood, 20 feet long, ballasted to float at the proper draft. In

the manufacture of a model, there are used plans known as " lines," the most important of which is the body plan—a drawing upon which are superposed the contour lines of sections cut by transverse vertical planes spaced along the longitudinal axis. With a pantograph the lines as received are converted to the model scale and wooden templates with wooden battens nailed to them are set up at the proper places on a bedplate, thus making a temporary form of the ship. Meanwhile, the rough blank form from which the model is to be cut is made by gluing pieces of wood together, and is placed on a cutting machine. This machine consists of a pantograph frame carrying a cutter whose motion is controlled by a roller, which when moved over the temporary form transfers its motion to the cutter which duplicates the profile on the model. As the pantograph frame is moved longitudinally the entire surface of the model is cut. It is then fitted with towing fixtures and ballasted to float at designed draft.

The towing carriage is a massive structure rolling on trucks along the sides of the basin; its weight of some 80,000 pounds providing sufficient inertia to suppress tendencies toward sudden changes in speed. The carriage is moved by motors geared to the truck wheels. The function of the carriage is to tow the model at uniform speed and to measure in pounds the resistance offered by the water to the passage of the model. The resistance is measured by means of a graphical recorder which shows the speed and pounds resistance at that speed. A series of runs is made at various speeds and usually at three displacements. The resistances thus obtained enable the resistance of the full sized ship to be obtained as indicated above.

The problem of resistance involves a great many variables so that even for a single vessel of given displacement and speed it would be a great undertaking to investigate the effect of changes in the dimensions and in the shapes of the water lines and sections. However, by assuming a definite shape or set of parent lines and deducing all models from these lines by variations of dimensions and coefficients of fineness, the problem can be enormously simplified. This has been done by Admiral Taylor with a series of eighty models which he calls the Standard Series. These models involved two values of beam-draft ratio, five values of displace-

ment length ratio, and eight values of longitudinal coefficient (*i.e.*, ratio of the volume of displacement of a vessel and the volume of a cylinder of section the same as the midship section and of length the same as the length of the vessel).

Although it would require experiments with models derived from an infinite number of parent forms to trace the effect of all possible variations of ships, this is unnecessary as long as the major factors involving resistance can be determined.

Trials at the Model Basin of over a thousand models, including the Standard Series, have justified the following conclusions as to the principal factors affecting the resistance of ships, not of abnormal form, and the relative importance of these factors:—

With a given displacement, speed, and frictional quality of surface, the skin, or frictional resistance varies with the length.

The principal factors having to do with residuary resistance are:—

1. The length.
2. The area of midship section.
3. The ratio between beam and draft.
4. The shape of midship section.
5. The details of shape toward the extremities of the vessel.

The effect of variations in these factors has been obtained experimentally and the resistance of the models tabulated. These resistances enable the effective horsepower to be obtained at once. Knowledge of the effective horsepower and the shape of the vessel enables the Bureau of Engineering to design a propeller, and with its efficiency known, to determine the power of the propulsive machinery to be used.

SPRING STYLES.

Armed with these Model Basin data and with preliminary machinery weights obtained from Engineering, the Bureau of Construction and Repair can advise the General Board that tentative characteristics may be laid down. With these characteristics in hand, sketch designs are made which will form the basis of further discussion. These sketch designs are made by taking the nearest type design and expanding it (usually) as necessary. Data of

ships in commission facilitate this. When a vessel has been completed very detailed reports are made of actual weights installed. These weights are divided into hull, hull fittings, armor, armament, engineering, and electrical weights. The displacement of the new ship is assumed and the weight under each heading is roughly estimated. If the sum of the weights does not equal the assumed total displacement modifications are made. Small scale plans known throughout the Navy as "spring styles" are made, showing the arrangement of the principal spaces in the vessel, such as the engine and boiler rooms, fuel oil tanks, turrets, magazines, etc. Heretofore, there has not been a great variety in these styles, as an approximate displacement was sufficient. With the Treaty for the Limitation of Armaments, however, the displacement of allowed vessels is absolutely fixed, and for light cruisers the size of guns was limited to 8 inches. A great variety of combinations on a given displacement can be devised, involving speed, armor, and guns. Before the characteristics of the *Pensacola* and the *Salt Lake City* were decided upon more than forty carefully worked out preliminary designs, all on a displacement of 10,000 tons were made, and "spring styles" submitted to the General Board. As the ideas of the Board crystallized several complete plans on a larger scale were made, models run to make sure of the power requirements, and the characteristics finally fixed and promulgated to the various Bureaus. This procedure is unusual and is due to the treaty limitation of displacement but it will be typical of future construction.

BUREAU OF ENGINEERING.

To enable the Bureau of Construction and Repair to prepare its plans, much information has been given it by the Bureau of Engineering which designs the propulsion machinery and its auxiliaries. The basis of the design for the propelling machinery is the effective horsepower ascertanied by the Bureau of Construction and Repair. The data obtained in the Model Basin are plotted on cross section paper—speed versus power—and three curves of effective horsepower, one for each displacement of the model are furnished to the Bureau of Engineering.

The screw propeller has been the subject of a tremendous amount of experiment and calculation. Supposedly an ancient Chinese device, it received little attention for ship propulsion until after the introduction of the steam engine. With the beginning of the 19th century, screw propellers of various designs were proposed in great numbers. The drawings of many of these shown in Bourne's classic "Treatise on the Screw Propeller" are very interesting. The subject has proved an attractive field for the inventor, the attraction apparently varying in inverse ratio to the knowledge of the inventor, and directly with his distance from deep water. However, Froude and Barnaby in England, and Taylor and Dyson in this country have greatly advanced the knowledge of screw propellers and the effect of hull design upon their performance.

Three principal methods of propeller design have been used:—

1. By "Direct Comparison"—when all the conditions for a satisfactory vessel of a form similar to the one under consideration are known.

2. By methods based on trials of model propellers in model tanks.

3. By methods based on actual trials of full-sized propellers driving hulls over carefully measured courses.

The third method is used by the Bureau of Engineering today. It is based on a 25-year study by Rear Admiral Dyson of trial trip data which cover the runs of a great variety of vessels of known dimensions, made in deep water, with clean bottoms. The revolutions of the propellers and the shaft horsepower are measured accurately. The enormous amount of data thus obtained has been analyzed and plotted into empirical formulae and curves from which may be obtained the propeller revolutions and shaft horsepower necessary to give any vessel the speed desired.

In deciding upon the propeller characteristics there are certain limitations which must be observed. The number of shafts to be used depends upon the power to be transmitted per shaft, the space limitations, and the chance to be taken on a complete breakdown of propelling machinery. Thus, while an old-time vessel with Scotch boilers and heavy reciprocating engines could be built with a single screw and steam along year after year reliably, although not very

economically, no combatant naval vessel of today is ever built with less than two screws—and our high speed cruisers as well as the electric drive battleships are fitted with four screws.

The location and maximum diameter of the propellers are fixed by the following condition:—

1. That the propeller wakes must be clear of each other.

2. That the shaft lines should not be inclined in any plane more than 1 1/2 degrees and must not be inclined more than 3 degrees.

3. That the propellers must not extend below the base line, *i.e.* the bottom of the ship, nor outside the extreme breadth of the vessel.

4. That they must have sufficient clearances between the blade tips and the hull of the ship, this clearance depending upon the draft.

These conditions quite definitely fix the diameter of the propeller. In determining the revolutions per minute for the propellers, the diameter, limited as described, as well as the type of vessel in question must be considered. Within certain limits, the propulsive efficiency of a propeller varies with the revolutions, greatest efficiency being found with slower revolutions. However, for modern high speed, high powered vessels where lightness is an essential, low speed of the propellers is not practical for several reasons. The power transmitted by the propeller varies with the square of its diameter, with its pitch, with its speed, and with the thrust per square inch.

The thrust varies with the 1.7 power of the projected area ratio, which is the relation of the projected area of the blades to the disc area of the propeller. The efficiency of propeller varies with this ratio, being a maximum when the ratio is 0.2. The efficiency drops from this until the ratio is 0.56, where it remains practically constant until a value of 0.725 for the ratio is reached. Tip speed of the blades (π DR) has been found to vary with the projected area ratio. The pitch of the propeller varies with the speed of the vessel, the number of revolutions and the slip. Pitch should be kept high, varying from 1.0 to 1.5 of the diameter of the propeller.

Analysis of a great amount of data shows that the efficient propeller has:—

(*a*) A low projected area ratio (0.2 to 0.56).
(*b*) Low thrust per square inch.
(*c*) Low tip speed.
(*d*) Low revolutions.
(*e*) High diameter.
(*f*) High pitch.

These ideals cannot always be reached due to the type of ship, its speed and power, the limiting diameter of the propeller and the type of machinery installed.

The shape of the after body of the vessel has much to do with the efficiency of the propeller also, as it determines how the water will flow to the propeller.

The diameter of the propeller is usually made as large as possible for all types of vessels—this more or less fixes the pitch. On electric drive battleships the revolutions can be made almost anything desired due to the reduction possible between turbine speed and motor speed. On the airplane carriers, due to the power transmitted, the revolutions found necessary were almost twice those of the *Maryland* class. On reciprocating steam engines, the limiting piston speed limits the revolutions possible. On destroyers and light cruisers with geared turbines, the revolutions must be kept high to keep down the size of shafting and the size of gear reduction.

Revolutions in the upper direction are limited by cavitation. Cavitation of a propeller is the condition occurring when the acceleration of water to the propeller is so rapid that the pressure falls below the amount necessary to preserve the natural flow of water to the screw.

It may be seen from the above that the design of a propeller is quite complicated and that any one propeller is a compromise of a great many factors. Before a propeller is finally decided upon perhaps a dozen have been designed, varying the diameter, the revolutions per minute, and the ratio of projected area to disc area to obtain the most efficient screw possible. Propellers designed by the Dyson method and installed on our latest vessels have shown an extremely close agreement between the estimated and trial performances. Such an agreement is a marked contrast to past

methods where as many as twenty sets of propellers have been tried in the effort to obtain the designed speed of a vessel.

Knowing the number of shafts to be installed, the shaft horsepower and revolutions of the propellers, and the steam pressure and superheat to be used in the boilers, inquiries are addressed to the various manufacturers to ascertain the water rates they will guarantee at the powers necessary for various speeds, as well as the approximate weight of the turbine units. It is just as necessary to know the steam consumption of a turbine developing power for a speed of fifteen knots as it is at full speed, for a low water rate at cruising speed may be more important than one at full power. The steam consumption of the main turbines (the product of water rate and shaft horsepower) is of course the largest part of the load on the boilers. The water rate of the turbines is, to a large extent, dependent upon the pressure and superheat of the steam supplied and of the vacuum which can be obtained in the condensers. The trend ashore today in central stations is toward higher steam pressures and superheats, and this trend is being followed in ships, though more slowly owing to the conservatism of sea-goers and the necessity of overcoming difficulties inherent in ships. Vacuums, closely approaching the limit, are now possible with the design of improved condensers and air removal apparatus.

Knowing the steam consumption of the main turbines at full power, an estimate of the steam necessary for auxiliaries is made. This is usually done by referring to past performances of a similar type of vessel using the same percentage for auxiliary steam. This includes ship's light and power generators, steam driven pumps, feed water and fuel oil heaters, etc. The total steam consumption is thus determined closely enough to estimate on the boilers. The type of ship under consideration determines the type of boilers to be used. Where space and weight are prime considerations as on high speed, high powered vessels, the express type of water tube boiler has been accepted as standard, and it is becoming the standard for all types of vessels. This type of boiler is being built for ships in units as large as twelve thousand square feet of heating surface. Steam can be generated very rapidly and the boiler can be forced to extremely high rates of combustion while maintaining an efficiency as high as seventy-five per cent. These qualities in a

boiler are highly desirable on a destroyer or light cruiser where the ratio of pounds of water evaporated per pound of installed boiler weight is a criterion, as well as the pounds of water evaporated per pound of oil burned. Battleships 49-54 which were scrapped had express type boilers contemplated. On steam driven auxiliary vessels where the power is comparatively low and forcing is unnecessary, large tube boilers of the water-tube type are usually specified.

The number of boilers projected depends upon the type of vessel and the power to be developed as well as the subdivision of units considered best. On destroyers four boilers down the center line seemed the best arrangement possible. On the airplane carriers the largest boilers available were used in the design and a sufficient number installed to give the necessary power.

Boiler manufacturers are given the steam pressure and superheat to be used, the number of boilers required, and the space available for each. They are asked to furnish plans, weights, efficiencies, and amount of heating surface of the boilers they consider suitable to generate the amount of steam specified. The plans and data received from the manufacturers enable engineering to ascertain what space is necessary for firerooms. The efficiency given by the manufacturer is a measure of the amount of oil to be burned, which in turn determines the size of the fuel oil pumps and the fuel oil heaters. The amount of oil to be burned also determines the capacity and air pressure of the forced draft blowers.

The total amount of steam to be used determines the capacity of the main feed pumps and feed heaters. The vacuum required on the main turbines, together with the steam consumption, determines the size of the main condenser with its injection and overboard piping, the main circulating pumps and the vacuum apparatus.

The type of main drive selected and its power determine the amount of lubricating oil required, from which is computed the size of the oil coolers, the oil cooler circulating pumps, the forced lubrication pumps, and the oil storage tanks.

Estimates of the electric load for all purposes are obtained by adding to the lighting and power load of engineering that necessary for the electrical machinery of other Bureaus in order to determine the size of the turbo-generators and their auxiliaries. The esti-

mates at this early stage are but approximations, inasmuch as the number of lights to be used can not be accurately determined until the lighting circuits are actually laid out, and power estimates by other Bureaus vary with changes in machinery design and compartment layout. However, the primary estimates allow the size of the ship's light and power generators to be approximated.

A great many details such as the size of smoke pipes, uptakes, safety valves, relief valves, steam and exhaust piping, water piping, oil piping, etc., must be calculated. These calculations are based on design formulae which limit steam and water velocities, friction drop in fuel oil piping, pounds of oil per square foot of stack area, and the like.

The refrigerating plant and the distilling plant are among the last things to be calculated as they are usually in isolated compartments and do not interfere with the main propelling plant. The refrigerating plant installed depends upon the size of the crew and the length of time for which fresh food must be carried. The evaporators are designed to furnish plenty of water for the crew as well as makeup feed for the propelling plant. Distilling plants operate on low pressure steam with a vacuum on the distilling condensers so that the auxiliary exhaust may be used to best advantage. Where fresh water must be obtained from sea water there is no more efficient way to utilize steam than to condense it in an evaporator, as the latent heat of the steam is there recovered instead of being lost in condenser circulating water.

The number of auxiliaries to be used on a given vessel is largely a matter of the judgment of the designer. After the total capacity of the auxiliaries has been determined it must be decided how to split up this capacity. The division of the units which the auxiliaries serve has a bearing on this, as well as the importance of the auxiliary, and the cruising requirements of the vessel. As most of the life of a vessel is at cruising speeds, auxiliaries efficient at the low power necessary for this speed would be very desirable. However, cruising speed requires but one-tenth to one-twentieth of the power necessary for full speed and the subdivision of auxiliaries cannot be carried to the point where those running will operate at maximum efficiency. The necessities of cruising speed are being met in some cases by the installation of special electric driven aux-

iliaries which have the benefit of the high efficiency of the electric generator.

Where the auxiliaries are steam driven and all auxiliary exhaust is utilized in the distilling plant and feed heaters, the loss due to auxiliaries running at low efficiency is more apparent than real anyway, as most of the heat lost in the machine is recovered when the steam is condensed. Where possible, important auxiliaries are duplicated, that is, there is one hundred per cent spare capacity installed. Where weight saving is paramount this cannot be done and the total capacity, broken into several units, is just sufficient for full power. A casualty to one unit means a loss of speed. However, the ratio between power and speed is not direct—a loss of half power on a ship like the *Omaha,* for instance means a loss of only 5 or 6 knots in speed.

After the design calculation has been completed a detailed estimate of the steam consumption is made. This is usually done for four speeds and for the "at anchor" condition. One of the speeds for which the steam consumption is calculated will be the endurance speed laid down by the General Board. The oil per knot required at this speed will be used to estimate the total oil necessary to give the cruising radius specified in the characteristics of the vessel.

Previous designs made as above described, and weight data obtained from completed vessels enable the Bureau of Engineering to give the Bureau of Construction and Repair approximate data for its preliminary design of a new ship which will be very close to a carefully worked out new design.

BUREAU OF ORDNANCE.

The Bureau of Ordnance is always ahead of new construction with its designs of guns and mounts so that for preliminary designs, weights and dimensions can be supplied immediately. These designs are not merely paper designs. Sample guns have been made, and tested at the proving ground to insure satisfaction in service. Armor is tested in the same way.

The General Board finally comes to a decision as to the ship which most nearly meets its requirements in all respects and the characteristics previously mentioned are sent to all interested Bureaus.

FINISHED DESIGN.

With these characteristics in hand the material Bureaus—Construction and Repair, Engineering, Ordnance, Aeronautics and Navigation, can begin the work leading up to the finished design which will culminate in the contract plans and detailed ship specifications for the new vessel.

The Bureau of Construction and Repair proceeds to draw up large scale hull plans called general arrangement plans. These consist of plans of all decks, and the hold, inboard and outboard profiles, and sections of the ship at various frames. In laying out general arrangement plans, certain major features must be given primary consideration—features which to my mind determine whether or not a vessel will be able to fulfill her mission on the day of battle. A successful ship must be able to get to the line of battle and stay there until her fuel is exhausted, while making her designed speed; must be able to fire her guns rapidly and accurately until her ammunition is exhausted; and must have a crew whose fighting edge is always sharp, whose morale is at the highest pitch.

The first consideration, then, is the arrangement of propelling machinery compartments. This depends on the type of machinery selected—confined today to electric drive, geared turbines, reciprocating (oil) engines, or combinations of these. Electric drive provides the most flexible arrangement as it allows the main machinery to be placed where it will have the most protection and interfere least with the ammunition spaces, while maintaining the water-tight subdivision considered necessary. Geared turbines are practically a necessity on high speed vessels such as cruisers or destroyers due to their lightness in comparison with the power developed. Oil engines heretofore have been confined to submarines or their tenders.

The arrangements of the lines of shafting must be carefully considered, particularly in ships of small beam where one engineroom is placed forward of the other. Boilers must be located so that there will be an easy exit for the gases to the uptakes and smoke pipes, which, in turn, must not interfere with other features of the vessel. The position of the machinery behind bulkheads, the oil

capacity and the underwater protection as well as the control of stability after a vessel is hit, all come into the question of getting a vessel to the fighting line and keeping her there.

The next consideration is the arrangement of turrets, guns, ammunition hoists and magazines. Turrets and guns must be disposed to give the maximum arcs of train and the minimum interferences with each other and with the ship's structure, insofar as blast effect is considered. The magazines must be well protected and so arranged that all ammunition can be conveyed to the guns as rapidly as needed. While Ordnance makes the layout of guns, mounts, elevating gear and turret ammunition hoists, Construction and Repair designs the turrets and broadside ammunition hoists. Ordnance must furnish information as to the type and size of the ammunition to be used by all guns, data as to the guns themselves, the type of mounts, requirements as to the ammunition supply and allowance, and all such data pertaining to the battery. Engineering enters the armament discussion because that Bureau lays out the size and arrangement of circuits for the electric motors operating the turrets and ammunition hoists, as well as for the fire control systems.

A general arrangement plan of the Bureau of Construction and Repair seems a far cry from the question of morale of the personnel of the vessel, but the forethought given to crew arrangements may easily determine whether a ship will be "happy," or a "mad-house." Nothing interferes with contentment as much as poor living conditions. On American vessels built for extensive cruising to all parts of the world, much thought is given to the amelioration of ship-board life.

A ship may be likened to a fortified city, able to withstand a siege for weeks at a time. In addition to this, business and pleasure must be carried on as usual and the city must be able to pick itself up complete and sail away at thirty miles an hour. This walled city is modern, too, with inhabitants who are accustomed to comfort and who have much business routine to carry out. Living conditions in a modern hotel or college dormitory must be duplicated as far as is possible. Rooms for officers have telephones, electric light, running water, air for ventilation, fans, comfortable beds,

desks and clothes stowage. The crew has the same except that many men live in one compartment instead of each having his own room.

The ship has its own public utility systems—electric light, ventilation, sewerage system, central heating plant, a fire and flooding system, fresh water supply (water which is distilled aboard), a telephone system, often of several hundred phones; a laundry, a printing office, which frequently prints a daily paper, a hospital and dispensary, a dentist's office, a cold storage and ice making plant. There is a butcher shop, a bakery, a tailor shop and ship's store where, on some ships at least, ice cream soda may be obtained. There is a garage (for motor boats principally, although an Admiral's automobile is sometimes carried from one port to another), a machine shop, blacksmith shop, carpenter shop, foundry, plumbers' shop and a post office. An open air theatre is provided by the upper deck where moving pictures are frequently shown, amateur theatricals held and boxing bouts staged. Large vessels carry Chaplains who hold divine service on Sunday. All vessels have a public library, a savings bank and a jail. An old time sailor man would be astounded at one of our modern ships.

After speed, guns and comfort have been cared for, the Bureau on Construction and Repair gives considerable weight to another feature—the appearance of the vessel. While in a way, the lines of a vessel have little to do with its fighting qualities, yet it is true that a well designed ship usually has a "right" look. A *"Flying Cloud"* with all sail set is productive of no greater thrill than a slim grey *Omaha* slipping along at thirty knots. To achieve a pleasing effect the masts and smoke pipes must be balanced, the sheer lines carefully drawn, and all the external elements of design properly combined. (Time alone will verify this as to the *Rodney* and the *Nelson,* "the ugliest ships ever built," and our own "one-sided" *Saratoga* and *Lexington.*)

PLANS.

General arrangement plans are usually made to the scale of one eighth inch equals a foot, so that considerable detail is possible. This detail is made possible by special studies on a much larger scale of important features of the vessel. For example, the anchor-

handling and steering gears are laid out very carefully, strength calculations being made of the principal parts to insure correct proportions. Ammunition stowage and handling arrangements must be worked out with great care. Engineering is given a plan of the spaces allocated for machinery and boilers, and from its calculations is soon able to tell if the space is sufficient. A body plan is made and a model to this plan is towed in the Model Basin, usually at speeds corresponding to half of the full speed of the ship and up to 2 or 3 knots above full speed.

While this first layout of the plans in detail is going forward, strength and weight studies are being made. These are based primarily upon the body plan but must be carried along with the plans as the latter develop.

The midship section and various other sections are drawn up with scantlings, that is, weights of plating, weights and types of angles and other shapes, based upon knowledge of previous practice and the construction of similar ships, increased or decreased in proportion to the increase or decrease in the size of the ship as compared to the type ship. With the limiting displacement now agreed upon, special studies are made as to the use of lighter material which will have the necessary strength. For each section the moment of inertia and the modulus of section are calculated. Then by means of the body plan and an approximate weight curve the bending moment at each section is obtained. The ship is assumed to be on a wave of her own length, first on a crest, then in a hollow, so that she is supported first in the middle, and then at the ends. Curves of buoyancy can thus be drawn and the difference between the curves at each section gives the bending moment at that section. Actually the calculations can be shortened greatly by various constants and type weight curves.

With the bending moments obtained at each section, the unit stresses in the deck and in the keel may be found, both in hogging and in sagging (crest and hollow). Generally it is found that in some sections the material will be strained beyond the allowed limit, and in others that the full strength of the material will not be developed. Then must be started a general redrawing of the sections, an increase of plating here, a decrease there, with possibly even a complete change in the extent of a deck or the position of a

bulkhead, until finally the stress curve is up to the allowed limit at all sections. The weight calculations run parallel to those for strength since the necessary scantlings for strength must be settled upon before they can be used to determine the weight. This weight calculation is made in considerable detail. As far as possible the material to be estimated upon is actually sketched out so that pure estimates from past practice may be eliminated. Curves are drawn showing the shell plating and frames expanded, the sections prepared for the strength calculations being used as a basis, and from these curves the most important groups of outside plating and of framing, the largest single items of structural weights in the whole problem, accurately calculated. For castings and large forgings, small sketches to scale are made, and the weight calculated from the volume.

While this work is going on in the Bureau of Construction and Repair, the other Bureaus are compiling information in regard to matters under their cognizance, and sending it in for incorporation in the plans. This information invariably makes changes in the originally contemplated layout of the vessel. New developments in machinery or service results on apparatus installed on ships in commission may change the type of machinery to be installed. Every effort, of course, is made to incorporate the latest developments into the new ship. Thus, Engineering may find out that forcing boilers to very high rates of combustion necessitates thicker insulation of the furnace walls. This in turn means a large boiler (for the same power), so that the boiler compartments will have to be enlarged. Ordnance may develop a new gun or a new type of ammunition and the whole arrangement of the battery may be changed. This involves a rearrangement of the magazines and hoists. There are always many changes from the first layouts, and now that a fixed displacement must be met, every change must be calculated to the last pound. When the report of the model test comes in it may appear that better results could be obtained with slight changes or with a modified bow or stern. In such cases the model may be altered and run again or a completely new model made. Finally it is agreed that the best results practicable have been obtained and the work can proceed. Actual scaled dimensions of various sections of the model as run are taken so that the lines of the ship

can be laid down in the Bureau, faired up, and eventually furnished the builders. Engineering checks its propeller design for various speeds, and the form of the ship with its power requirements may be said to be finally settled.

Another check on weights is made and this must be inside the authorized displacement with a considerable margin to allow for weights which invariably appear and which cannot be foreseen at this early stage of the design. If the weights plus the margin are beyond the allowed displacement, changes must be made either by the omission of some feature previously considered essential, or by the use of different material which will accomplish the purpose with less weight.

It can easily be seen that the design of a naval vessel is a process of cut and try, a compromise of a number of conflicting interests. However, compromise in the design of a new vessel is usually accomplished without much friction, and plans and data sent to the Secretary of the Navy for approval. With this approval in hand, the work of preparing the contract plans and the specifications is commenced. These plans and specifications form the basis for the contractors' bids and will also form the basis of the contract for the construction of the vessel.

CONTRACT PLANS.

Contract plans for a new vessel are prepared by the Bureaus of Construction and Repair and Engineering. These plans consist in general of the lines, which are laid down, faired up, and traced from the data furnished by the Model Basin; the midship section which shows in the greatest detail the construction of the ship, with the scantlings of the different parts of the structure, the type of riveting, and the details of special features of construction; the outboard and inboard profiles, and the general arrangement plans of deck, sometimes called joiner plans, which show the general layout of the ship on each deck and in the hold. The above plans are prepared in the Bureau of Construction and Repair. The Bureau of Engineering's contract plans generally show in as much detail as possible, the arrangement and location of the boilers, main engines, propellers, shafting, all auxiliaries and the steam, exhaust and feed water piping.

In addition to these contract plans there are a great many others known as type plans. These consist of compartment and access plans which show for each deck the types of doors, hatches, scuttles, ladders, etc., to be fitted; ventilation plans, water and oil systems; turret and gun foundations, armor and armor fittings; layout of electrical systems, fittings, switchboards, plotting rooms, bridges, fire control stations, etc.

SPECIFICATIONS.

To accompany these plans are the detailed ship specifications—one volume prepared by the Bureau of Construction and Repair for the hull, hull fittings and ordnance outfit, and another prepared by the Bureau of Engineering, covering the boilers, propelling machinery, auxiliaries and electrical plant. These special detail specifications cover the dimensions of the vessel, the materials to be used, thickness of plating, size of frames and longitudinals, type and size of boilers, machinery, auxiliaries and electric plant—in general the specific information the shipbuilder needs for the vessel in question.

Reference is made constantly in the detail specifications to the "General Specifications." There are two volumes of these, one covering the hull and ordnance material, and the other the propelling machinery, boilers, auxiliaries and electric plant. In these specifications are put down in great detail the requirements which must be met for everything going into a ship. For instance, under "Boilers, Oil burning, Express Type" is put down the material of the drums and tubes, the thickness and kind of refractory material for the furnace wall and bottoms, insulation of the boiler casings, type of oil burners, their spacing from each other and from the tubes and brickwork. With this information all that is necessary in the special specifications is the number of boilers, amount of heating surface, pressure, and degree of superheat.

Behind the General Specifications are the "Leaflet Specifications." There are nearly two thousand of these specifications covering the sixty-five classes of materials into which naval stores and material have been divided. The index runs from "Abrasive Wheels" to "Zinc, slab (spelter)" and includes almost everything that goes aboard ship either as a complete article or as a material

fabricated into machinery. For instance, there are twenty-two separate specifications covering steel for various purposes. These specifications are issued as Navy Department Specifications and are available to anyone who wishes them. Frequent mention is made of them in the General Specifications, and a list of the index numbers is published in the detail specifications.

CONTRACT WITH SHIPBUILDER.

An estimate of the cost of the completed vessel has been made by the material Bureaus and furnished to the Secretary of the Navy who submits this estimate to Congress when recommending a building program. When Congress has authorized the building of the vessel and appropriated the money for its construction, the final steps can be taken in the Navy Department. Contract plans are signed by the Secretary of the Navy and by the Engineer-in-Chief and the Chief Constructor. A "Circular of Requirements" and a proposed form of contract are prepared by the Judge Advocate General's office in consultation with the Bureaus. The "Circular of Requirements" gives extracts from the naval act authorizing the construction of the vessel, and states when the bids will be opened. The form of contract includes the price, time and place of delivery, agreements as to changes, claims for damages, design, material and workmanship; guarantees as to weight, economy and performance of machinery; trials of the vessel, and many legal points included to safeguard both the Government's and the contractor's interests. The contract plans, specifications, circular of requirements, and form of contract are sent to those shipbuilders who have indicated that they will bid on the vessel. On the date set for the opening bids, they are brought from the post-office by an official of the Judge Advocate General's office, and in the presence, usually, of the Secretary of the Navy, the Chiefs of Bureau of Construction and Repair, and Engineering, and representatives of the bidders, opened for examination. The principal figures are read aloud and the papers are turned over to the two Bureau Chiefs for analysis and recommendation. The work of analysis is very carefully made by the design and financial sections of the two Bureaus, as not only must the plans and specifications submitted be

carefully studied on account of changes and exceptions inserted, but also the guaranteed performance of the vessel taken in connection with the price quoted and the proposals as to payment and penalties must be examined. A joint recommendation as to award of contract is made by the two Bureau Chiefs to the Secretary of the Navy who signs the contract with the successful bidder. Upon signing of the contract, corrected copies of all contract and type plans and the completed detail specifications are sent to the contractor.

CONSTRUCTION OF THE VESSEL.

At the same time copies of the contract, type plans and detail specifications are sent to the Inspector of Machinery, the Inspector of Ordnance, and the Superintending Constructor at the works of the shipbuilder. These officers with their assistants maintain close supervision over the construction of the vessel and the installation of machinery until it is delivered to the Government.

As soon as the contract is signed, the shipbuilder commences the hundreds of plans necessary for the estimates of material needed and its fabrication and installation in the vessel. All material purchased by the contractor is inspected by the Inspectors of Naval Material whose districts cover the whole of the country. Machinery is tested at the factory for workmanship and performance. All plans drawn by the shipbuilder are scrutinized at his works by the Inspectors, and important ones are forwarded to the Bureaus for approval before actual construction proceeds.

The steps involved in the building of a vessel from the data furnished by the Bureaus are as follows:—

Drawing of detail plans from contract and type plans furnished;

Ordering the material;

Making of the molds, templates, patterns, etc.;

Fabrication of the material;

Assembly of material into the vessel.

After the detail plans are made which enable the shipbuilder to tell what he needs, the material is ordered from various manufacturers. The greater part of this material is, of course, steel for the hull. The plates and shapes present the greatest difficulty, inasmuch as the size of each must be specified so as to allow for

shaping and cutting to the proper size without excessive waste. The shell plates, keel stringers, etc., are usually ordered from a wooden model made to scale. Frames, reverse frames and similar parts which must be bent are measured from the plans.

The making of molds, patterns and templates commences as soon as the plans are complete, even before the material is received. Plans for castings are sent to the pattern shop if they are to be made in the shipbuilder's foundry, or to the outside concern who may make them.

In the mold loft the lines of the ship are laid down to full size and faired up. Then on a special flooring called a scrieve board is drawn a complete body plan, and the lines cut in with a sharp knife. From this board is obtained the data for templates, which in addition to the shape of the piece, carry data as to the location of rivet holes, edges to be sheared or planed, lightening holes, etc. Templates are made for the shell plating, deck plating, bulkheads, inner bottom plating, frames, reverse frames, floors, keels, keelsons, deck-beams, bulkhead stiffeners, brackets, bounding bars, coamings, etc. Special machines have been developed for the preparation of all of these parts so that they may conform exactly to the templates.

Before assembly of the material commences the building ways, or slip, have been prepared. It is customary to strengthen the ground over which a vessel is to be built with piling so that the concentrated weight of sometimes thousands of tons may be adequately supported. On top of the piles are placed cross logs securely bolted in place. Along the center line of the ship are laid large blocks from twelve to eighteen inches square, upon which is laid the keel of the ship. These blocks are built up high enough to give the workmen ready access to the bottom and to give sufficient room for launching. The line of the keel blocks is inclined usually with a slope of a little more than half an inch per foot. (The larger the vessel, the less the declivity necessary.) The launching ways are usually located so that they are about one third the beam of the ship apart. The size of the launching ways depends upon the launching weight of the vessel and is usually such that the bearing pressure is from two to two and one half tons per square foot. They must extend some distance beyond the edge of the

water so that there will be several feet of water over the end of the ways when the vessel is launched. Certain calculations must be made as to the behavior of the vessel when launched to make sure that the launching ways have been properly designed. These calculations must be made before construction of the vessel commences as the driving of piles, dredging, etc., must be done before laying the keel.

When the building ways are ready and sufficient fabricated material on hand the keel is laid, usually with some ceremony, as this marks the actual commencement of construction. The work proceeds as rapidly as possible, as time is, of course, a factor in the ultimate cost and delay beyond the contract date of completion entails a penalty. When the underwater shell plating has been riveted and calked and the progress of construction of the ship otherwise is considered sufficiently advanced, she is launched. The amount of work done previous to launching varies within wide limits. Merchant vessels have been practically completed on the building ways, ready to get up steam and move, when launched. In other cases the launching weight may be the determining factor. Again, if the ways are not needed as soon as possible, the hull may be practically completed. A ship is easier to work on when on the ways as everything is rigid and the ways are frequently built with a roof to protect the workmen.

In any case, before a ship is launched she must have sufficient work done to give her the requisite buoyancy, strength and stability for flotation. The weight and position of all parts going into the ship on the ways are carefully noted so that the longitudinal and vertical centers of gravity may be calculated. Sometimes it is necessary to ballast a ship to give her sufficient metacentric height to insure her stability. This is seldom necessary in the case of naval vessels as enough engineering weights usually have been installed to keep the center of gravity down. Before launching many underwater fittings are installed, such as the rudder, struts, propeller shafts and propellers, bilge and docking keels, etc.

The launching ways are ordinarily built with a greater declivity than the building ways, having about three quarters of an inch more slope to the foot. Launching ways are made in two parts—the ground ways and the sliding ways. Shortly before the time set for

launching, the ground ways are laid in position, blocked up, and secured to prevent spreading. Their upper surfaces are coated with launching grease, and the sliding ways, with their lower surfaces similarly greased, are laid in position. The cribbing which is to transmit the weight of the hull to the sliding ways is installed loosely, the wedges between it and the sliding ways being not driven home. The above procedure is carried out as short a time as possible before the launching in order to preserve the grease.

Carefully prepared arrangements must be carried out on the day of launching in order that the ship may take the water at the desired time of day (usually depending upon the state of the tide). A large gang of men is detailed for driving home wedges, knocking out keel blocks, and they must know exactly what to do and when to do it. Wedging up is done simultaneously and at such a time that the vessel will be borne by the ways for as short a time as possible before the launching. When the vessel is raised clear of the keel blocks they are knocked out, leaving the vessel ready to launch. Launching is accomplished by various forms of releasing devices (hydraulic pistons, triggers, or the cutting of retaining timbers). As the vessel slides down the ways, her momentum must be checked in order that she will not run too far in the usually restricted waters near the shipbuilders' plants. This is sometimes a considerable problem as in the case of the airplane carriers, which had a launching weight of some twenty-five thousand tons.

The launching of a naval vessel is always the occasion of some considerable ceremony, the vessel being christened by a sponsor chosen because of some connection with the name selected for the ship.

After launching the remaining work is completed as rapidly as possible. During the entire construction of the vessel the plans and workmanship are under the closest possible supervision of the Bureaus' Inspectors who do not hesitate to hold up work being done improperly or not in accordance with the contract.

Tests are made of boilers, generators, and auxiliaries as they are installed, and finally the machinery is given a dock trial. Sometimes a shipbuilder takes a ship out in free route for a builders' trial before announcing her ready.

TRIALS.

When the vessel and her machinery are substantially complete except such minor items of work as may, in the discretion of the Secretary of the Navy, be left until after the trials, and the contractor has made sufficient trials to be reasonably sure of the vessel meeting the contract, the Secretary is notified by the contractor that he is ready to submit her for trials.

The Board of Inspection and Survey is ordered by the Department to attend these trials for the purpose of making an examination of the vessel and her equipment and of witnessing and reporting upon the performances of the vessel and her machinery. The Board is furnished with copies of the contract and specifications for the construction of the vessel and is guided in the performance of its duties by the contract, plans and specifications and duly authorized changes therein and by such specific instructions as may be included in the precept. As the responsibility for the success of the trials rests with the contractor, the Board has no control over the vessel or machinery but merely over the instruments for recording data, and the contractors are not to be interfered with in the management of the vessel or machinery.

Usually there have been ordered to the works of the shipbuilder some time before the completion of the vessel a number of officers and enlisted men who will be attached to her when commissioned. These include the Captain, Executive Officer, Engineer Officer and other Heads of Departments and many of the leading enlisted men. Some of these had been made assistants to the Inspectors and Superintendent Constructor and so were able to have supervision over the installation and tests of machinery they would later operate.

The Board, during the trials, makes such observations as will enable it to make a full and detailed report to the Department. This report includes the following:—

Statements from the Superintending Constructor and Inspector of Machinery that the vessel and her machinery are in accordance with the contract, plans and specifications;

Lists of unfinished work;

Reports of inspections of bottom, fuel tanks, reserve fuel tanks, chain lockers, double bottoms, etc., which cannot be inspected by the Board at the time of trials;

Trials of anchor windlass, rudder, boat cranes, winches and capstans;

A statement as to the horsepower developed, accompanied by standardization data, curves of speed, power and revolutions of the propellers, and of water and fuel consumptions, and a synopsis of the trial;

A statement as to the weights installed in the vessel. These weights are divided into a number of groups which facilitate analysis and assist estimates for future designs;

The opinion of the Board as to the working of all parts of the machinery and whether or not the performance is satisfactory.

The object of the report is to give the Department the fullest possible knowledge of the condition and performance of the vessel, and nothing is omitted which is conducive to that end.

The machinery trials are specified in the contract and usually consist of the following:—

Standardization trials;—these are made over "measured miles" of which the Government has several. The one off Rockland, Maine, is usually used for large, high-speed vessels built on the Atlantic coast as the depth of the water and its temperature meets the requirements for full power running and for condenser injection water.

The trials consist of three runs, alternating in direction, for a number of specified speeds, usually commencing at ten knots. At the contract full power speed five runs are made. During all of these runs the time necessary to cross the range, the total number of revolutions of the engines, and the indicated or shaft horsepower are carefully measured. Special electrical devices are put aboard for ascertaining the exact time of each run and the total number of revolutions turned up during this time. The power is measured by taking indicator cards on reciprocating engines, by torsion meters in the case of turbine driven ships, or by electrical measurements on electric drive ships. The data of the runs at each speed are averaged, thus getting rid of the effect of the tide. From these data can be plotted the usual curves of power versus speed and revolutions versus speed. A standardization run astern with the machinery developing its maximum backing power is also frequently included.

With the standardization data available, the fuel and water economy runs can be made. These trials vary, of course, with the type of vessel concerned, but a sufficient number is made to give a fair curve of the fuel and water consumption, and to test the machinery in various combinations. These tests vary in length from four to twelve hours. A light cruiser is tested at two or three different speeds up to the limit of speed of the cruising combination, and then up to the contract horsepower with the high speed connection, and with usually a short test at the maximum power available. On electric driven vessels runs are made with the various combinations made possible by several generators and different motor windings. Further trials are made of the ability of the vessel when going ahead at full speed to back satisfactorily at full backing power, and of her ability to go ahead satisfactorily at full ahead power when under full stern way.

The fuel oil and water consumption of the boilers and machinery are generally measured today by carefully calibrated meters, as the installation of measuring tanks with the attendant piping system is expensive and cumbersome.

After the trials are completed an examination is made of such parts of the machinery as would show injury if any occurred, or defective conditions if any is likely to be present.

As soon as the report of the trials is completed, the contractor is required to make good defects which have occurred and to complete the unfinished work as soon as possible in order that the vessel may be delivered to the Government. Usually a vessel is delivered to the Government with some items uncompleted. These can be completed when the vessel is fitting out, usually at the Navy Yard to which she is delivered. Upon delivery of the vessel she is considered preliminarily accepted and usually is commissioned immediately. The officers and crew have been assembled, organization completed, and when certain equipment and stores are put aboard the ship is ready to sail on her " shake down" cruise.

On this cruise the crew and ship become acquainted with each other. The vessel's maneuvering qualities are ascertained, tactical data are taken to find out her turning circle with various rudder angles, how much way she carries when approaching a dock or buoy, etc. Each member of the crew finds his niche, the engines

and boilers are tried out at all speeds, the guns are fired at various angles of train and elevation, and the vessel, from a mass of iron and steel, becomes almost a living entity.

The six months after preliminary acceptance is the guarantee period and any weakness, deficiency, defect, failure, breaking down or deterioration that may occur, other than that due to fair wear and tear, shall be repaired to the satisfaction of the Department at the expense of the contractor. At the end of the six months' period the final trials are made.

These trials are identical with those made on preliminary acceptance, except that they are carried out by the ship's force. The Board of Inspection and Survey witnesses these trials as it did the preliminary trials and collects data for a final report.

And so ends the period of perhaps six years from the first hearings before the General Board to the time when the Trial Board can report that the vessel is strong, well built, and in accordance with the contract, drawings, and specifications.

CONDENSER TUBE FAILURES.

By H. G. Eberhart, Lieut., U. S. N., Member.

There are few casualties in a marine engineering plant that cause more trouble or annoyance than condenser tube failures. It is true that repairs can be effected and the ship enabled to resume station, but, it is far from pleasant to find that a tube leak has developed.

Aside from the inconvenience of repairing the damage caused by tube failures, the loss occasioned by penalties or low scores made while running on one engine render it impossible for a ship to stand well up in the competition if condenser casualties are frequent. It is possible, also, that severe tube leakage may endanger the safety of the vessel if the failure occurs in a heavy storm near a lee shore or in any other situation where it is imperative that full maneuvering ability be maintained.

When the stage of deterioration is reached that failures are so frequent as to render the performance of the vessel uncertain and examination shows the tubes to be in a generally poor condition, the only lasting remedy is a partial or complete retubing of the condenser in question. This is an expensive navy yard job and in these days of limited funds should not be undertaken too hastily.

The natural life of condenser tubes varies greatly. Some condensers have failed completely in a year or two; others last from ten to fifteen years. Tube life depends upon many factors, such as quality and velocity of the circulating water, impingement on the entrance edges of the tube, turbulence of the circulating water in the entrance chamber, manner of securing the tubes, and their chemical and physical composition. These are questions of design and manufacture. The operating engineer has no control over them but it should be borne in mind that the natural life of the tube may be materially shortened by lack of care on his part. This may or may not show up in his tour of duty but it will appear eventually. Some operating engineers have, in the past, been prone to regard tube failures as a necessary evil and so awaited the time

for a leak to start with a good supply of soft pine plugs and the hope it wouldn't happen, or, if the condition of the condenser was bad, started an intensive campaign to have it retubed.

It seems unlikely that of two ships of the same type, made by the same firm, of the same age, and operating under the same conditions, it should be necessary to retube the condensers of one in about 3 years while the condensers of the other were in good condition, yet, such has been the case. The natural assumption is that there must have been some difference in the care given these units. There have, however, been condenser tube failures that could not be traced to any lack of care on the part of the operating engineer.

The following discussion will endeavor to explain the types and causes of condenser tube failure. These are of two general classes, viz.:

(1) The tube cracks or splits.
(2) The tube corrodes.

Failures by cracking and splitting are the least common but are likely to be most sudden and severe. There is no gradual and prolonged rise in salinity as in the case of a small leak. Occasionally the split is so large that enough sea water will be taken in to cause the feed tank to overflow. There are two distinct types of failures of this nature, namely:

(*a*) The straight longitudinal split with sharply defined edges. This is known as " Season cracking."

(*b*) The irregular, fibrous looking crack which may run in any or several directions.

Season cracking is due to faulty manufacture. The specifications for finished tubes call for a certain strength in compression. The manufacturer, knowing the hardening effect of " cold work" after the final anneal, gives the tube a cold finishing pass to avoid rejection on account of lack of strength. This cold pass stresses the tube as shown in Figure 1.

This shows that the stress is most intense at the inner and outer surfaces of the tube wall where the tube is in contact with the rolls and mandrel. The intensity of the stress depends upon the degree of cold work given. If it is of such intensity as to stress the surface of the metal beyond the elastic limit, the amorphous cement,

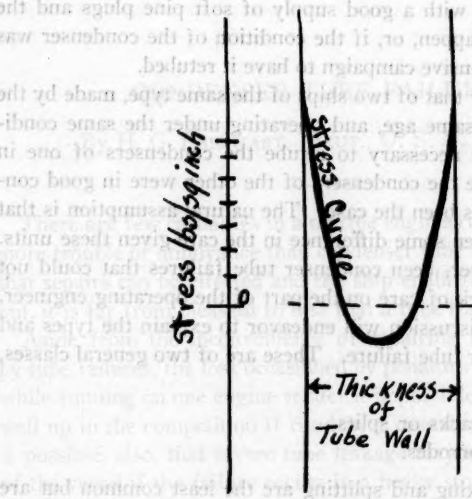


FIGURE I.

a substance between the crystals, fails and the tube splits under the action of the remaining internal stresses or upon application of small strains received in service.

The remedy is to anneal the tube at a temperature below the recrystalizing temperature after the final pass. Once the tube is installed it is too late. The recommended annealing temperature is shown in Figure II.

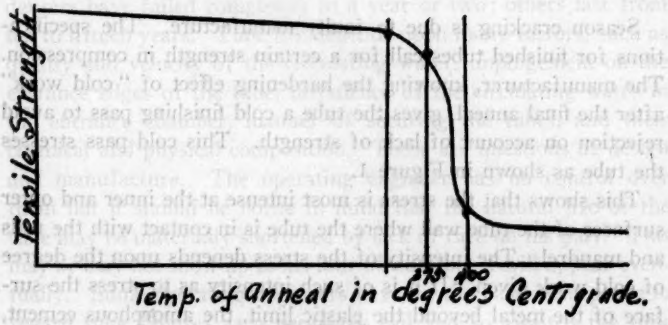


FIGURE II.

This figure is a graphical illustration of the results various investigators have obtained from tests conducted with a view of ascertaining the relationship existing between the temperature of the anneal and the tensile strength of the tube after annealing.

It will be seen that up to 350 degrees C. (662 degrees F.) there is practically no change in tensile strength. After 375 degrees C. (707 degrees F.) tensile strength rapidly decreases to a minimum at 400 degrees C. (752 degrees F.). After this temperature is passed there is little change. Since the reason for annealing is to remove the effect of cold work, it should be applied in such a manner as to obtain the maximum effect without an undue loss of strength in tension and compression. The diagram shows that up to a temperature of 375 degrees C. (707 degrees F.) there is but little loss of strength. If the temperature is carried higher, strength rapidly falls off and at 400 degrees C. (752 degrees F.) the loss of strength is excessive due to the fact that this is the melting point of zinc. As a result of these investigations it is generally conceded that 375 degrees C. (707 degrees F.) is the correct annealing temperature for condenser tubes. This is indicated by a bluish color of the tube surface. A red heat is too high as it carries the metal to a point of low tensile strength.

It is fortunate that the Inspector of Material has a simple and positive means of detecting tubes that are liable to develop season cracks. The test consists merely in dipping the end of the tube in a 1.5 per cent solution of mercuric chloride. Following this test, "Season cracks" will develop within four hours if the tube has been incorrectly finished.

The irregular, fibrous appearing crack is caused by "dirty" metal containing non-metallic inclusions or oxides. This condition can be detected by a microscopic examination of the etched surface. Non-metallic inclusions will stand out clearly because of their black appearance and total lack of resemblance to the crystals comprising the metal. The oxides are recognized by their dissimilarity of shape and color as compared to the grains of the metal. There is no remedy for this condition and its existence is solely the fault of the manufacturer. This type of failure is of rare occurrence.

The more common type of failure that necessitates most of the retubing of condensers, is failure by corrosion. In order to discuss

this action it will be necessary to consider the composition of the tube in relation to the copper-zinc Equilibrium Diagram.

Condenser tubes are usually made of a ternary alloy of approximately the following composition, copper 70 per cent, zinc 29 per cent, tin 1 per cent. This is the formula for Admiralty metal and produces an excellent tube if properly made. Copper is very resistant to corrosion, but a tube of pure copper would lack the necessary strength. To produce this quality, zinc is added in limited quantities. An excess of zinc would cause brittleness and for reasons shown later, renders the tube less resistant to corrosion. Due to the formation of a hard brittle substance of undetermined composition, the addition of tin materially increases the strength and hardness of brass. Since it has the effect of decreasing the ductility of the metal, tin should be kept below 2 per cent. It has been found by experience that a small amount of tin increases the resistance of the metal to corrosion by sea water. The amount of tin added is so small that from this point in the discussion the metal may be considered as a binary alloy of copper and zinc. In explanation, the alloy will be referred to the Cu-Zn Equilibrium Diagram, Figure III.

It will be noted that the Equilibrium Diagram has temperature as an ordinate and percentage composition as an abscissa. It is interpreted as follows:— Assume a mixture of copper 90 per cent and zinc 10 per cent heated to 1100 degrees C. At this point the entire mass will be in a liquid condition since it is above the temperature of the liquidus line. Upon cooling to about 1050 degrees C. the mass will consist of alpha crystals in liquid. Upon further cooling the temperature will fall below that of the solidus line and the entire mass will be solid and consist entirely of alpha crystals. It should be noted in this connection that once the solidus is passed there are no further transformations to take place and therefore, the metal will be composed of alpha crystals. These alpha crystals are what is known as a solid solution, *i.e.*, they appear to be composed of but one constituent similar in appearance to a pure metal. This point is of special significance as will be shown later.

Now if a metal of 60 per cent Cu., 40 per cent Zn., is assumed, it will, if cooled slowly, be a solid at about 950 degrees centigrade and will be composed of crystals of alpha and beta. Upon fur-

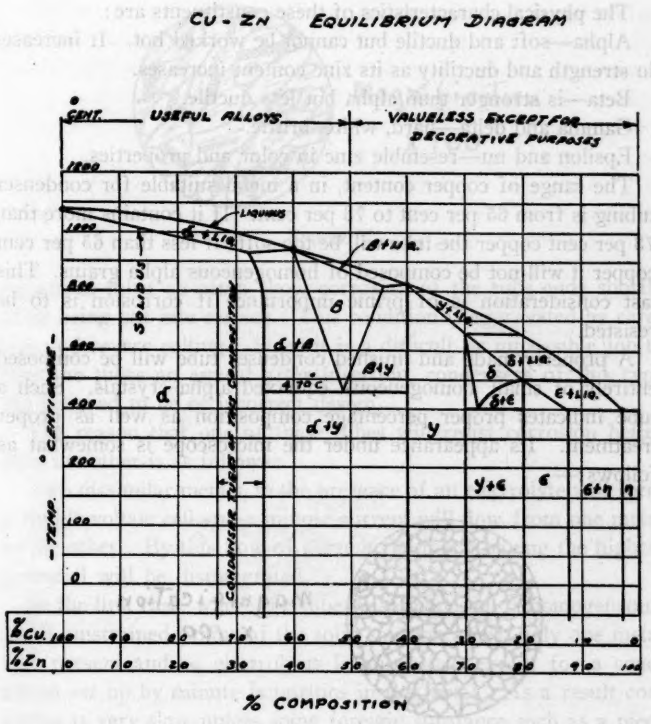


FIGURE III.

ther slow cooling the metal will undergo a transformation at 470 degrees C. and will then be composed of crystals of alpha and gamma. If on the other hand the metal is quenched from above 470 degrees C. the transformation does not have time to occur and the result is a metal composed of alpha and beta grains. A metal composed of alpha and beta is soft while one of alpha and gamma is hard. A metal of this composition is annealed by quenching from above 470 degrees C. by virtue of the fact that by so doing, the gamma constituent does not have time to form.

As the zinc content increases we find other constituents appearing; *i.e.*, delta, epsilon, and nu.

The physical characteristics of these constituents are:

Alpha—soft and ductile but cannot be worked hot. It increases in strength and ductility as its zinc content increases.

Beta—is stronger than alpha but less ductile.

Gamma and delta—hard, white, brittle.

Epsilon and nu—resemble zinc in color and properties.

The range of copper content, in a metal suitable for condenser tubing is from 65 per cent to 73 per cent. If it contains more than 73 per cent copper the tube will be too soft; if less than 65 per cent copper it will not be composed of homogeneous alpha grains. This last consideration is of prime importance if corrosion is to be resisted.

A properly made and finished condenser tube will be composed entirely of small, homogeneous, equiaxed alpha crystals. Such a tube indicates proper percentage composition as well as proper treatment. Its appearance under the microscope is somewhat as follows:—

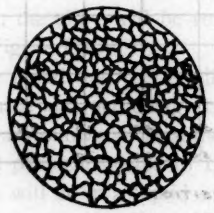


FIGURE IV.

Such a tube is very resistant to corrosion.

If now the same tube had been improperly finished, *i.e.*, cold worked after the final anneal either by passing through rolls, bending for insertion or rolling into a tube sheet, the grains in the locality of the cold work will be distorted as shown in the sketch below: This shows the grains to be distorted and stretched as a result of the cold work. The grains furthermore, appear to be covered with fine scratches. These are known as slip lines. Some of them also show the banded appearance known as "twining." These are always indicative of cold work. This type of tube will corrode quickly and offers an explanation for the fact that condensers with

FIGURE V.

expanded tubes so often show corrosion of the tube ends shortly after being put into service. This condition is aggravated by careless and severe rolling. Since it is a difficult or impossible job to give the tubes an anneal after insertion, condensers of this type are believed to be of incorrect design.

The reason that one of these tubes will resist corrosion better than the other is as follows:

Two dissimilar metals, in the presence of an electrolyte will form a simple voltaic cell and a minute current will flow from one metal to the other. By this flow of current the metal having the highest potential will be disintegrated.

In the first case, where the tube was composed of homogeneous, small, unstrained grains of the solid solution alpha, only one metal was present and so electrolysis is impossible except for a small action set up by minute impurities in the metal. As a result corrosion is very slow unless some foreigin substance such as a piece of coal should lodge in the tube. If this should occur the other element would be supplied and local action would occur. This shows the necessity of keeping foreign matter out of the tubes.

In the second case where the grains are strained, there is, by virtue of that condition, an electrolytic dissimilarity between grains, very much as though they were of different metals. As a result, electrolytic action occurs at a rapid rate as soon as the electrolyte in the form of sea water is supplied.

This is the explanation and right or wrong is an attempt to answer an observed fact. That is, observers in the metallurgical field have found that the cold worked tube is much more subject to corrosion. For this reason the expanded tube when in contact

with salt water is wrong in theory. It should further be considered that tubes of this type in fixed heads are continually expanding and contracting and therefore, bending. This keeps them subjected to cold work throughout their entire life.

The other phase of the corrosion problem is in the case of the tube being of the wrong chemical composition.

By looking at the Equilibrium Diagram it is seen that the line dividing the alpha region from the alpha plus beta region is very sharply defined. If a tube is made of a composition containing slightly too much zinc, the alloy will be composed of crystals of alpha surrounded by envelopes of beta. The effect of this is of having two dissimilar metals in contact. These, in the presence of sea water, form the voltaic cell previously described and corrosion starts.

There are two stages by which the corrosion goes to completion, viz.:

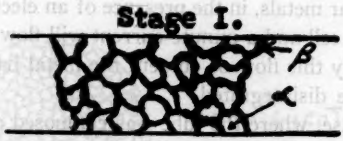


FIGURE VI.

This shows the typical structure at the beginning of the process, *i.e.*, bright colored alpha grains surrounded by dark beta boundaries. When the electrolyte comes in contact with the surface of the tube, beta becomes the anode and alpha the cathode. Dezincification occurs; a little metallic copper is left behind and Stage two results.

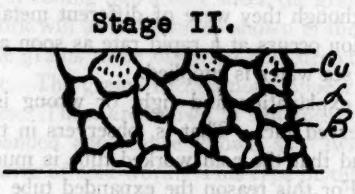


FIGURE VII.

In this stage the pure copper becomes the cathode, the alpha constituent the anode, and electrolysis continues until the tube disintegrates unless active measures are taken. This type of corrosion is easily recognized by the copper-like appearance of the fracture when the tube is broken or the presence of pure moss copper in the pits. Such a case when viewed under the microscope will have the following appearance:

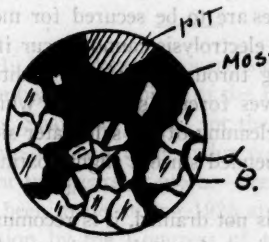


FIGURE VIII.

This shows how the porous moss copper forms at the surface, and how, as the alpha disintegrates by electrolysis, the wall is penetrated, leaving soft, spongy regions through which salt water can seep. This action may be general as at tube ends or it may be extremely local in the form of pits.

From the foregoing discussion it is seen that in most cases the manufacturer, or the designer is at fault. This does not mean, however, that the operating engineer can do nothing about it. Having been given an installation to take care of, it is up to him to do his utmost to prevent corrosion. Against failures by splitting or cracking he can do little.

Since corrosion is admittedly due to electrolytic action, it is logical and correct to expect the installation of zinc plates to reduce the damage to the tubes to a minimum. In this connection the condition of the zinc plates should be kept in mind. They must be kept free from scale and should have good metallic contact with the condenser head. If this is done the zinc will be disintegrated instead of the tube. It is no great amount of trouble to examine zincs once a week. This procedure also offers the advantage of frequent inspections of the tube ends. In this connection, it is rec-

ommended in the case of tubes expanded into the tube sheets, that the ends be frequently coated with boiled linseed oil. This protects the ends, somewhat, from erosion and corrosion.

In the discussion, so far, we have considered only the part played by the tube. Another factor is the electrolyte. Since time is a function of electrolytic action it is believed that in the case of expanded tubes, at least, the condensers should be drained of sea water and the tubes cleared by blowing through with compressed air whenever the main engines are to be secured for more than a few days. It is evident that electrolysis cannot occur if no electrolyte is present. The blowing through by air, in addition to clearing the tube of water, removes foreign substances which may set up local action. Excessive cleaning of the salt water side of the tubes by cutters is not recommended since a slight coating of scale acts as a protective agent.

In case the condenser is not drained, it is recommended that the circulators be run daily in order to keep the condenser clear of stagnant water. This applies to augmenter, distilling and refrigerating condensers as well. In many localities condenser tubing disintegrates rapidly because of certain inanimate organic substances such as sewage, etc., in the circulating water. The acids resulting from their decomposition attack the tubes and corrosion is accelerated. Sea water is not generally so affected but it does contain much live organic matter. If the water is confined as in an idle condenser these minute organisms die and decompose. Corrosion, as a result, is accelerated.

It is believed that, from an understanding of the principles involved and the observance of the remedies suggested, condenser tube corrosion can be greatly reduced. Even after tube failures have started, the number and frequency of casualties due to this cause can be reduced by an intensive campaign along the lines set forth. Any reduction of tube failure is, from a standpoint of economy, efficiency and peace of mind, well worth the effort expended on care and preservation.

BUILDING GUNBOATS IN CHINA.

Description and Trials of U. S. S. "Luzon," "Mindanao," "Panay," and "Oahu."

By Commander Bryson Bruce, U. S. Navy, Member.

On March 1, 1926, contracts were signed for the construction of six river gunboats at the plant of the Kiangnan Dock and Engineering Works, Shanghai, China, for the U. S. Navy. The building of these vessels had been under consideration by the Navy Department for two years or more. Preliminary design of a 180-foot boat had been made during 1925 and upon authorization and the appropriation by the Congress of the necessary funds, bids were requested from builders in Asiatic waters who were in a position to build and deliver the vessels at Shanghai. On account of the light construction required it was impossible to run these vessels any distance in the open sea. Bidding was, therefore, considered to be impracticable except by firms at Shanghai or Hongkong. It was believed possible to send the boats along the coast from Hongkong to Shanghai and since being constructed two of the vessels have made this voyage without great difficulty.

The successful bidder, the Kiangnan Dock and Engineering Works, is a Chinese government yard located at Shanghai above the city on the Shanghai side of the Whangpoo River. Offices of the U. S. Government representatives were opened promptly after the signing of the contracts at the works of the Company and the preliminary work started. The steel material for the hulls of the vessels was required by U. S. law to be of American manufacture. Orders for this material were placed promptly by the contractor; however, delivery was not effected as soon as expected and it soon became evident that a delay in final completion would be caused on this account. China at this time was concerned with a revolution and it also was expected that delays probably would occur on account of unsettled conditions.

It might be well to state something of the conditions in China at the time of the signing of the contracts in order to better undersand the troubles with which the Chinese officials of the dock yard had to contend in carrying out their contracts. On March 1, 1926, the local political conditions at Shanghai were good but not stable. The province of Kiangsu, of which Shanghai is the principal city was under the sway of Marshal Sun together with several adjoining provinces. He was the so-called war-lord in power at the time.

It very soon became evident to the Inspectors for the U. S. Navy that the Kiangnan Dock and Engineering Works, both the Chinese Officials of the dock yard and their foreign advisors, desired to live up to every clause of the contracts, to every item of the specifications. They were anxious in every way to meet the wishes of the Inspectors and remained so under very trying conditions. Conditions in fact that at times became so bad that less resolute men might easily have given up in despair.

On account of steel not arriving, no keels were laid until the fall of 1926. Ways had been prepared for all six vesels, planning for construction to proceed on the hulls simultaneously. Keels were finally laid

No. 43 U. S. S. *Guam* on October 17, 1926.
No. 44 U. S. S. *Tutuila* on October 17, 1926.
No. 45 U. S. S. *Panay* on December 18, 1926.
No. 46 U. S. S. *Oahu* on December 18, 1926.
No. 47 U. S. S. *Luzon* on November 20, 1926.
No. 48 U. S. S. *Mindanao* on November 20, 1926.

In the meantime, construction had been going ahead on the main engines, main condensers, and some other items for the *Guam* and *Tutuila* at the dockyard. Other machinery to be furnished by the contractor was ordered, certain materials were being assembled by the Bureaus of the Navy Department and shipped to Shanghai, and plans for the main engines for numbers 45 to 48 inclusive had been forwarded to the Bureau of Engineering, Navy Department, Washington, for the construction of the main engines in America. Prior to awarding contracts, it did not appear that the main engines for the first two vessels to be completed—*Guam* and *Tutuila*—could be manufactured and delivered at Shanghai without holding

U. S. S. "MINDANAO" IN GORGES OF THE YANGTSE RIVER.

up the completion of these vessels. This assumption would have proved true if construction could have proceeded without any interference.

Sometime in the fall of 1926, the Chinese General Chiang Kai Shek started north from Canton with the avowed intention of conquering all of China. At first his movements caused no concern, his continuous victories, however, soon brought him to the territory under the control of our local war lord, Marshal Sun, then nominally under the Peking government, which government, being the recognized government of China, had signed the gunboat contracts.

Chiang Kai Shek did not stop at the borders of Marshal Sun's domain but steadily forged on, in advance came unrest among the citizens which was plainly evident among the workmen at the dockyard. Conditions grew gradually worse, work less effective, the control of the authorities insecure, until early in March of 1927, Marshal Sun lost control of the Shanghai district and it was taken over by Marshal Chang Chung Chang, war lord of the neighboring province of Shantung. He was represented locally by Admiral General Pi. At this time conditions at the dockyard were demoralized and it was impossible for the authorities to accomplish anything. The Commander-in-Chief of the Chinese Navy had gone over to the Canton side about this time. Hoisting the Cantonese flag on his ships, he pulled out for Woosung at the juncture of the Yangtze and Whangpoo Rivers to await the taking of Shanghai by Chiang Kai Shek. During this time, agitators were busy exhorting the workmen to various demands and making more and more headway. No one could stop it.

On March 21, 1927, the Cantonese or Nationalists under General Chiang Kai Shek took charge of the dock yard under considerable confusion, some fighting, with minor casualties. For several weeks no progress was made in the building of the gunboats. The Directors of the dock yard who were serving under appointment of the Peking government were not molested, conditions gradually improved, until within a few months conditions were again normal—for China.

There were no further disturbances of this nature, the boats were advanced one by one to the launching stage, launched, the machinery installed and other work completed.

Each vessel as completed was given trials as required by the contracts. The results of these trials with a description of the vessels in the case of the *Guam* and the *Tutuila,* the first two vessels to be completed, were published in the August, 1928, issue of the JOURNAL.

Launching dates were as follows:

No. 43 *Guam,* May 28, 1927.
No. 44 *Tutuila,* June 14, 1927.
No. 45 *Panay,* November 11, 1927.
No. 46 *Oahu,* November 26, 1927.
No. 47 *Luzon,* September 12, 1927.
No. 48 *Mindanao,* September 28, 1927.

In order that certain data may be available for comparison, it will be given for the *Guam* class as well as for the other two classes. The *Guam* and the *Tutuila* were sister ships and will here-

GENERAL DATA.

	GUAM Class	PANAY Class	LUZON Class
Length between perpendiculars, feet and inches	150-00	180-00	198-00
Length overall, feet and inches	159-03.25	191-00	210-07.5
Length on L.W.L., feet & inches	149-09	180-00	198-00
Beam, moulded, feet and inches	27-00	28-00	31-00
Beam on L.W.L., feet and inches	27-00.125	28-00.375	31-00.5
Beam, extreme, feet and inches	27-11	28-11.375	32-01.5
Draft, designed, fresh water feet and inches	5-03	5-06	6-00
Displacement on designed draft, tons	387	474	615
Displacement per inch on L.W.L. tons	7.55	9.2	10.94
Coefficient of fineness, block	.656	.615	.60
Coefficient of fineness, midship section	.991	.986	.977
Coefficient of fineness, L.W.L.	.801	.785	.605
At designed draft, area immersed midship section, square feet	140.67	152	180
Area L.W.L. plane, sq.ft.	3250	3960	3720
Area wetted surface, sq.ft.	4260	5052	6198
Gravity tank, capacity, gallons	150	150	150
Ship's tanks, fresh water, capacity, gallons	2040	2020	2492
Reserve feed tank, capacity, tons	7.12	7.78	7.79
Engine room feed tank, capacity, gallons	100	100	100
Fuel oil to meet designed cruising radius, tons	37.2 *	35.5	38.3
Fuel oil bunker capacity (95 per cent full), tons	77.48	110.08	168.39

* Two boilers in use at ten knots.

after be referred to as the *Guam* class, the *Panay* and the *Oahu* as the *Panay* class, the *Luzon* and the *Mindanao* as the *Luzon* class.

The above table shows the differences in some of the main characteristics of the three classes. The *Guam* class was described in some detail in the August, 1928, issue of this JOURNAL and will not be repeated here.

The *Panay* class, being thirty feet longer than the *Guam* class, have this additional length to include features and additional accommodations not possible in the *Guam* class.

The Captain's quarters consisting of cabin and bath are located on the upper deck, also one officer's stateroom, as in the *Guam* class. On the main deck forward there are four officer's staterooms, one of them double, the wardroom, pantry and officers' bath and toilet,—this is two more staterooms than in the *Guam* class. The general arrangement is the same as the *Guam* class for all quarters. The Chief Petty Officers have a room in the after deck house on the upper deck containing eight bunks, necessary lockers, and space for messing. The crew are quartered in the after deck house, bunks being provided for forty with lockers and messing arrangements. There are seven built-in berths in the hold on the starboard side forward for mess attendants, and space for native boat crews in the after hold.

This class have two separate firerooms with one boiler in each room as in the *Luzon* class, the *Guam* class having only one fireroom. There is a door at main deck level between the two firerooms, the forward fireroom is entered from the main deck on the port side through an air lock, the entrance to the after fireroom is from the starboard side. Some space and weight was saved in the *Guam* class with the single fireroom with two boilers facing each other. Otherwise, the *Panay* class is very much the same as the *Luzon* class which will be described in more detail.

The general data of the hull of the *Luzon* class is as given above. They are twin screw open stern vessels with three rudders, constructed with the lightest weight of material consistent with sufficient strength for river service. All structural steel plates, except bullet proof, and all iron and steel fittings are galvanized by the hot dip process.

The construction and general arrangement of these two vessels is very much the same as the *Guam* class described in detail in the August, 1928, issue of this JOURNAL. This class is larger, of higher power and speed and are built as flagships. There is not much difference in the size of the crew, so that the increase in space is largely used for machinery and flag quarters, with some addition to compartments used for other purposes.

There are some marked differences in internal arrangement as compared to the *Guam* class and rather than to enter into full description of this class in detail, only the main points of difference will be mentioned, it being assumed that, otherwise, they are practically identical with the *Guam* class.

The *Guam* class had two boilers in one fireroom with fuel oil tanks outboard on each side of the fireroom. These vessels have two separate firerooms and, in addition to oil tanks outboard, have a tank which extends across the ship just forward of the forward fireroom.

On the main deck there is not a great deal of difference except in quarters for officers and the absence of a galley. There are seven officers' staterooms on this deck, one of them double, and a wardroom twice the size of the *Guam* wardroom, together with baths and pantry.

The flag quarters are on the upper deck, also the Captain's quarters, one officer's stateroom, and flag office. The Captain's cabin is just aft of the bridge, he does not have a private bath, and from the after bulkhead of his stateroom at from 22 1/2 to frame 37 are situated: starboard side, flag office, 3 1/2 frames; port side, flag bath and toilet, 3 1/2 frames; aft of these, flag stateroom, 3 1/2 frames; then flag cabin, 4 frames; then an officer's stateroom on starboard side and flag pantry on port side, 3 1/2 frames. Over these quarters is a "Palm-garden," for the use of the flag officer, fitted with grass rugs and wicker furniture. It is covered with double awnings and makes a delightful place for use during warm weather.

The general galley is located on this deck between the stacks, whereas on the *Guam* and the *Panay* classes it is on the main deck. The Radio room is over the engine room on this deck instead of forward of smoke pipes as on the *Guam* and the *Panay* classes.

Otherwise the three classes are very much the same as regards internal arrangement.

There are two masts, two smoke pipes, no fireroom but two engine room ventilators.

COMPLEMENT.

Officers	Flag		Ship	
	Admiral	1	Line	3
	Staff	4	Medical	1
		Total		9

Men	C. P. O.	Other Ratings
	4	45
	Total	49

There are accommodations for eleven officers, one room doubled. The chief petty officers' quarters has berths for eight; crews' quarters, 53; servants' quarters, 7; and, in addition, a compartment aft for Chinese boatmen.

The galley, fresh water system, pilot house, steering gear, anchor engine, capstans, fire and sanitary systems, sick bay, and awnings are very much the same as the other classes. The galley only differs in location, the pilot houses and sick bays are identical. Other items differ as to the size and detail arrangement but are the same in general installation and type.

BOATS.

The *Luzon* and the *Mindanao* have two 26-foot motor sampans, instead of one as the other classes, and one 22-foot pulling sampan. The motor sampans are powered with Model 50 engines manufactured by the Kermath Manufacturing Company, Detroit. They accommodate 10 persons, are of light draft—about 13 inches—have a nineteen-inch propeller placed in a tunnel and show a speed of approximately nine knots over a measured mile. The hull is of teakwood with galvanized steel frames. The cockpit is forward and covered with a canvas automobile type top with permanent glass windshield with doors for entrance. It has the usual boat equipment.

STEERING GEAR.

The steering gear is similar in all three classes. The three rudders have a total area of 97.89 square feet and the steering engine cylinders are of 5-inch bore.

ORDNANCE.

There are two 3-inch, 50 caliber, anti-aircraft guns, with bullet proof shields, one located on the upper deck, centerline, forward of the bridge, and the other aft of the sick bay on the centerline. The forward gun has an arc of fire from dead ahead to 115 degrees aft on each side, and the after gun from dead astern to 140 degrees forward on each side. There are eight machine gun mounts with bullet-proof shields installed on the upper deck, a nest of three on each side at about the forward smoke pipe, and one on each side aft near the main mast. There is the usual equipment of small arms and ordnance accessories including a one-meter range finder.

THE "LUZON" AND "MINDANAO" MACHINERY INSTALLATION REQUIREMENTS.

The contracts required a speed of not less than 16 knots over a measured mile and a fuel capacity to give a steaming radius of 1200 miles at 10 knots. The draft with two-thirds supply of fuel, lubricating oil, water, and consumable stores was not to be greater than 6 feet. It should, therefore, be noted that a steaming radius of 800 miles on the draft of 6 feet would fulfill contract requirements, providing bunker fuel capacity was sufficient for 1200 miles. There was a severe penalty for failure to make the speed, and also a penalty for overdraft and failure to meet the cruising radius requirement. The boiler installation was to produce the steam necessary for 16 knots for main engines and other uses while burning not over .5 pound of fuel, 19,000 B.T.U.'s per pound, per square foot of heating surface. There was no penalty for failure to meet this requirement.

The design was for 16 knots at 310 R.P.M. and 3150 I.H.P. The trial results were 16 knots at 288 R.P.M. and 1795 I.H.P. for the *Luzon* and 292 R.P.M. and 1718 I.H.P. for the *Mindanao*. The steaming radius required was 1200 miles at 10 knots. The

bunkers of each vessel hold 168.39 tons of fuel oil at standard gravity and temperature. In the case of the *Luzon* the trial results indicate that it requires 71.53 tons per 1200 miles leaving a possible reserve supply of 96.86 tons, and in the case of the *Mindanao* the corresponding figures are 46.07 tons and 122.32 tons. The maximum speed and corresponding I.H.P. were: *Luzon,* 18.29 knots and 3560 I.H.P.; *Mindanao,* 18.15 knots and 3668 I.H.P.

It will be seen from the above that the speed was exceeded by over 2 knots, the required cruising radius was equalled with 96.86 tons of reserve fuel in the case of the *Luzon* and 122.32 tons in the case of the *Mindanao.* The *Luzon* 10-knot trial was with two boilers, while only one was used on the *Mindanao* trials, accounting for the apparent difference in cruising radius with full fuel supply. There is no real difference.

ARRANGEMENT OF MACHINERY.

The machinery arrangement is in general the same as for all three classes, except that there are two separate firerooms with a door between them at the main deck level on the *Panay* and *Luzon* classes, the *Guam* class having but one fireroom.

OPERATION UNDERWAY.

Underway there is no special condition except there is only one main condenser, one main air pump, and one main circulating pump. It is possible to put the general service pumps on the condenser and to drain it—with the atmospheric valve open—to the reserve feed tank. Generally underway both boilers will be in use, or at least in the gorges as a precautionary measure.

OPERATION IN PORT.

A feature of port operation, which means a considerable saving in fuel, is the installation of a Ray Rotary oil burner, electric driven, on each boiler. With this burner, which is supplied with oil direct by a small electric driven pump set, it is possible to operate with an open fireroom and unnecessary to use the forced draft blowers, steam fuel oil service pumps and fuel oil heaters. In weather where steam heat is necessary and with a supply of fresh water, fires may be died out in both boilers, the 10 K.W.

kerosene generating set operated and the vessel can accomplish all tasks usually requiring power except to pump up the gravity tank. The refrigerating machinery, machine tools, radio, ventilating fans, lights, and all electric appliances may be operated including the electric driven flushing pump—not all at one time, but with care, the generator need not be overloaded.

MACHINERY INSTALLATION FURNISHED IN PART BY GOVERNMENT.

The contracts required a part of the machinery installation, including the manufacture of the main engines to plans furnished by the contractor, together with a few other items to be furnished to the Kiangnan Dock and Engineering Works by the Government, and to be installed by them under the supervision of the Government representatives. The items furnished in this manner were: main engines, turbo-generator sets, motor generators for interior communication, kerosene generator set, switchboard, all electric wire, fixtures, junction boxes, etc., for the complete electrical installation, searchlights, ventilating fans, signal systems, radio equipment, galley equipment, foam fire extinguishers, engine revolution counters, machine tool equipment, bullet proof steel, all the ordnance outfit, together with spare parts for the above and the complete allowance of other equipment and supplies. The above was forwarded to the works of the contractor who assumed responsibility for its care and safe-keeping from the time of delivery until acceptance of delivery of the completed vessel.

MAIN ENGINES.

The main engines were built at the Navy Yard, New York, from plans furnished by the contractor, tested and shipped assembled to Shanghai. There are two engines, triple expansion, vertical inverted, three cylinders arranged in order and with diameters as follows: H.P., 15 inches; M.P., 23 inches; L.P., 35.25 inches. The stroke is 18 inches and the designed R.P.M., 310.

The engines are built on a bedplate consisting of two steel channel bars carrying cast steel transverse girders for support of main bearings and the forged steel engine columns. The thrust block is carried by these steel channel bars also and the whole is securely bolted to the engine foundations.

Cylinders are cast separately of close grained cast iron and are firmly secured to each other with faced joints and fitted bolts. The H.P. cylinder has a cast iron liner. The H.P. and M.P. valves are piston type with cast iron liners, the L.P. valve a double ported balanced slide valve. Cylinders are fitted with relief valves, the M.P. and L.P. with impulse steam, and all cylinders with indicator gear and drains. All cylinders are covered and lagged with 85 per cent magnesia lagging protected by polished steel sheet secured by brass screws.

Pistons are of forged steel and fitted with snap rings held in place by a steel follower. Piston rods are of forged steel with crossheads forged on the bottoms. The brass shoes, lined with white metal, are fitted to crossheads and are adjustable for wear. Connecting rods are of forged steel.

Valve gear is of the Stephenson link type with adjustable expansion blocks for each cylinder.

Eccentrics are of cast iron keyed to the shaft. Straps are of bronze lined with white metal with liners between halves for adjustment. Rods are of forged steel with fork end at the top fitted with gunmetal brasses.

The reversing engine is fitted at back of engine with reversing lever located at throttle. Engines are fitted with hand reversing and jacking gear.

The crank shaft is a built up forged steel shaft with three cranks 120 degrees apart.

The thrust block is a steel casting fitted between channel bars of engine bedplate. It has a bearing at each end and five collars on the shaft with five cast steel white metal lined horseshoes, each arranged to be adjusted independently and with water cooling.

Main bearings are bronze, lined with white metal, and are held in place by polished steel caps, steel bolts, nuts and locking pins. They are fitted with brass oil boxes with hinged covers.

Lubrication is by hand. All working parts are fitted with brass oil boxes and tubes as necessary.

Main bearings and guides are fitted with water service. The water supply is taken from main circulating pump or the general service pumps.

The engines are of light weight and high R.P.M.—360 possible —and need to be handled in service as a light duty engine. The parts are readily accessible for overhaul with good headroom over the cylinders with necessary lifting gear.

MAIN CONDENSER.

There is one main condenser of Uniflex type located on the center line between the shafts and aft of the engines. It has 1870 square feet of cooling surface, 1265 tubes 3/4 inch in diameter and 7 feet 11.25 inches long. Tubes are packed at both ends in rolled brass tube sheets. The shell is of steel and the heads of cast iron fitted with inspection plates. The delivery pipe from circulating pump is on the starboard side and is 14 inches in diameter; the overboard discharge is also on the starboard side. The main engines, the turbo-generators, circulating pump, air pump and steering engine exhaust directly to the main condenser. The turbo-generators also exhaust directly to the auxiliary condenser and the main air pump may be cut into the auxiliary exhaust. The auxiliary exhaust may be cut into the main condenser if desired.

MAIN AIR PUMP.

There is one main air pump between the engines forward of the main condenser. It takes suction from the condenser and discharges to the hot well between the main feed pumps located at the forward engine room bulkhead. It is of vertical simplex type, $14 \times 26 \times 15$, manufactured by G. and J. Weir. The suction is 7 inches and the discharge 6 inches in diameter.

MAIN CIRCULATING PUMP.

There is one main circulating pump of Kiangnan design and manufacture. It is a centrifugal pump driven by a single cylinder steam engine, located on the starboard side of the engine room near the main injection. The engine cylinder is of 5 1/2-inch bore and 7-inch stroke. It takes suction from overboard or bilge and discharges through main condenser overboard also to water service for main engines.

MAIN FEED PUMPS.

Two main feed pumps are located in engine room at forward bulkhead. They are of Weir design and manufacture, are of vertical simplex type, 10 1/2 × 7 1/2 × 16, with 3 1/2 suction from feed tank, reserve feed tank, and overboard, and 3-inch discharge to boilers via grease extractor and feed heater.

AUXILIARY MACHINERY IN ENGINE ROOM.

The following machinery is identical in size and design to similar machinery of *Guam* class; the auxiliary condenser, Worthington, 142 square feet, air and circulating pumps, 5 1/2 × 8 × 8 × 7; fresh water pump, Worthington, vertical simplex, 3 1/2 × 4 × 4; sanitary pump, same size as fresh water pump; general service pumps, two Worthington, vertical simplex, 6 1/2 × 7 × 8; electric driven sanitary, Gould, 3-inch bore, 2 1/2-inch stroke, motor silent chain drive; distilling plant, Weir high pressure, 200 gallons per hour.

TANKS.

There are ten fuel oil tanks, four on each side outboard of the boilers and two athwartship forward of the forward fireroom, having a capacity, when 95 per cent full, of 50,384 gallons, giving —at 300 gallons to a ton—about 168 tons. Two lubricating oil tanks on the port side of the engine room, one for engine oil, 135 gallons capacity, and one for forced feed oil, 57 gallons capacity. In the passage way on main deck aft of engine room and forward of crew's quarters is a 215-gallon tank for kerosene for use in kerosene generating set,—it is piped to a small tank near the engine. A tank located near the ice machine for oil holds 57 gallons; the ship's storage tanks, 2492 gallons; the reserve feed is 7.79 tons; and feed and filter tank, 150 gallons.

SHAFTING.

The line shafting and propeller shafts are of government specifications, Class B steel. They are solid shafts and propeller shafts are interchangeable between vessels and between sides of same vessel. A spare propeller shaft was furnished for each vessel.

PROPELLERS.

The propellers are out turning, solid cast, bronze, four blades, and manufactured by the contractor. The pitch is 7 feet and diameter 6 feet and 2 inches. A spare set of propellers was furnished for each vessel.

MISCELLANEOUS.

Lubrication of machinery is the same for all three classes.

The York 1-ton CO_2 refrigerating plant is the same as the *Guam* class except as to size of cold storage compartments which are slightly larger.

The repair equipment is identical for all three classes with the exception that the machine tools are located on the port side of engine room, whereas in the *Guam* and *Panay* classes they are on the starboard side.

The steam heating system is in general as in the *Guam* class, also the hot water heater.

The siren and whistle are of same type and location but of a larger size.

The Weir feed heater is similarly located but is of 125 square feet as compared with 90 square feet in *Guam* class.

The 5000 cubic foot ventilating fan for engine room is located in the starboard after corner of the engine room in front of the switchboard; this fan and the switchboard being on the port side in the *Guam* and the *Panay* classes.

BOILERS.

The two oil-burning boilers are located in separate firerooms with ample working space. They are of the Thornycroft design and manufacture, oil burning, express type, with one steam drum of 4 feet 2 inches diameter, and two water drums of 30 inches diameter. The heating surface is 4200 square feet each; 1966 cold drawn steel tubes of 1 1/8 inch diameter, 116 mils thick, the longest tube is 7 feet 1 inch; 168 cold drawn steel tubes of 1 1/2 inch diameter, 128 mils thick, the longest tube is 7 feet 5 15/16 inches. The furnace volume per boiler is 450 cubic feet. The working pressure is 250 pounds.

Boilers are fitted with two Klinger type water gauge glasses, automatic feed regulator, and all other usual fittings. The safety valves are fitted with easing gear which may be operated from outside the boiler room on the main deck. The steam and water drums are covered with from 2 to 3 inches of magnesia lagging, reinforced with chicken wire, and protected on the water drums with sheet metal and the steam drum by asbestos cloth.

Each boiler has its own uptake and smoke pipe, with manhole in uptakes, four handholes at base of outer smoke pipe, manholes in steam and water drums, and soot doors in casing for use in inspection and cleaning.

Each boiler has arranged on the front along the bottom four mechanical atomization Thornycroft oil burners, and in the center, over these burners an electric driven Ray Rotary oil burner for port use.

ARRANGEMENT OF THE FIREROOMS.

The firerooms are made air tight for closed operation. The oil tanks are arranged outboard on each side of the firerooms and athwartship forward of forward fireroom with high and low suction to each tank. The suctions may be closed from outside the firerooms on the main deck. There is one steam driven fuel oil service pump in each fireroom and takes suction through strainers from the supply main. Oil can be transferred from any starboard tank to any port tank or vice versa. These pumps discharge through duplicate strainers, oil meter with by-pass and duplicate oil heaters to burners. There are individual valves to each burner on burner manifold and a quick closing master valve in the supply line. There is a hand driven pump for use in lighting off.

The Ray Rotary burner is located in the center of the boiler front and is supplied with oil by its own pump set, a small rotary pump driven by a chain and motor. There is one Ray burner and one pump set per boiler. With this burner in use only, the fireroom is opened, and it is not necessary to use any of the main apparatus, only the Ray burner and small pump being in use. This pump discharges through a small oil meter of a size suitable to the amount of oil being used.

The firerooms are built of bullet proof plates where exposed to direct fire, and bullet proof plates also are in way of oil tanks.

There are the usual foam type fire extinguishers and steam smothering line is run around the firerooms close to oil tank bulkhead and also underneath the boilers.

The forced draft blowers are located at the height of the main deck, one in each fireroom, and take suction through a special bullet proof grating. There is a platform built across the firerooms at the height of the main deck with gratings along the sides of the boilers, which gives easy access to boiler fittings, fuel oil heaters, foam fire extinguishers, and lockers for stowage. The firerooms are entered from opposite sides through air lock doors at height of gratings, with ladders leading down to floor plates or by trunk leading to a second air lock door at floor plate level. There is communication by door between firerooms at main deck level.

MACHINERY AND EQUIPMENT IN EACH FIREROOM.

One Weir, horizontal, simplex fuel oil service pump, $6 \times 4 \times 8$.

One Kiangnan forced draft blower, single cylinder, steam driven, reciprocating, forced lubrication. The cylinder is 6 1/2 inches in diameter and 5 inches stroke. The fan is 6 feet 3 1/2 inches diameter.

One Weir oil heater of 95 square feet heating surface each.

Fuel oil suction and discharge strainers in duplicate.

One 1 1/4-inch Buffalo fuel oil meter in discharge line from main service pump, and one 1/2-inch Buffalo fuel oil meter in discharge line from pump sets for Ray burner.

One electric driven rotary fuel oil pump set, capacity 60 gallons per hour, 1/4 H.P. motor, chain drive, furnished by Ray Manufacturing Co.

One two-cylinder hand-driven pump is in the after fireroom only for oil supply in starting up, not needed if kerosene generator is in operation as Ray burners can be used.

One foam fire extinguisher, one gauge board, one vise, one manometer and eight-day clock.

ELECTRIC INSTALLATION.

The main generators, two Westinghouse, turbo-electric, 25 K.W.; one auxiliary generator, kerosene driven, 10 K.W.; the main switchboard and searchlights are identical with those of the *Guam* and *Panay* classes, and in general the alarm system, call

bells, lighting circuits, etc.; are the same except such minor differences as were necessary on account of the difference in size and arrangement of the vessels. The only exception of importance is the installation of a Western electric interphone system of twelve stations, and a battery charging panel. The battery charging panel is located in the engine room near the main switchboard and the batteries in a compartment underneath a bench in front of the switchboard. This compartment has ventilation at the bottom with an exhaust leading from the top to the outside atmosphere. The batteries are in a lead lined tray. This system is principally for the use of the flag officer and his staff.

In addition to the above telephones there are voice tubes from and to the principal places as on the *Guam* and *Panay* classes.

RADIO INSTALLATION.

The apparatus is identical with that of the *Guam* and *Panay* classes. The radio room being located in a different part of the ship and the masts being further apart, has made some difference in the installation of antennae. The motor generator is located in the engine room hatch underneath the radio room.

Data obtained on trials is given below for one vessel of each class, the data obtained from the other vessel of each class checking very closely.

	Guam	Panay	Luzon
Date of trials	Dec.15,1927	Aug.31,1928	May 22,1928
Speed knots	10.02	10.0	10.0
R.P.M. main engines	190.2	180.4	170.04
Slip of propeller, per cent	20.96	13.58	14.875
Displacement, tons	401.1	471.28	586.8
Displacement, normal tons	387	444	585
Corresponding draft, ft.& inches	5-03	5-02.75	5-09.94
I.H.P. main engines	267	277	342
Cutt off in decimals of stroke	.63-.76-.73	.65-.65-.65	.74-.74-.74
Main steam at boilers, lbs.gauge	249	250	250
" " " engines, " "	240	249	249
H.P.steam chest, pounds absolute	75.6	46.5	54.5
I.P. receiver, pounds absolute	29.2	15.0	21.1

	Guam	Panay	Luzon
L.P.receiver,lbs. absolute	4.1	7.5	9.0
Vacuum,inches	25.2	25.2	25.9
Main air pump. double strokes per min.	23	31	24
Main circulating pump, R.P.M.	193	166	166
Main feed pumps,double strokes per min.	9	5	8
Fire & bilge pump, " " " "	23	48	26
Main injection,temperature degrees F.	55	82	74
Overboard discharge, " " "	70	111	90
Outside air, " " "	53	81	66
Engine room, " " "	85	108	87
Boiler feed from heater " " "	164	218	217
Main feed pump discharge,pounds gauge	291	262	259
Turbo-generator,steam pounds gauge	245	255	244
volts	126	120	125
amperes	88.6	94	104
kilowatts	11.16	11.28	13.0
Number of boilers in use	2	1	2
" " burners in use per boiler	1	2	1
Square feet H.S.boilers in use total	5200	3000	8400
Fuel oil, pounds per hour	885.2	777.15	1355.36
" " " " I.H.P.main engines	3.32	2.8	3.9
B.T.U's per pound of fuel oil	19000	19000	19000
Fuel oil,lbs per square foot of H.S.	.17	.259	.159
Air pressure, firerooms,inches water	1.04	1.4	1.0
Boiler sq.ft. of H.S. per I.H.P.main engines	19.48	10.83	24.56

	Guam	Panay	Luzon
Fuel oil service pumps, double strokes per min.	6	3.8	4
Fuel oil to heater, temperature degrees F.	85	103	94
" " from " " " " "	143	180	199
" " specific gravity	.9353	.9356	.9353
Fireroom temperature above gratings, degrees F.	125	152	120
" " floor plates " "	108	124	115
Pressure fuel oil to burners, lbs. gauge	120	107	50
H.P.M. blowers	298	322	198

It should be noted that in the above trials of the *Guam* and *Luzon* two boilers were in use, while only one boiler was used on the Panay.

Principal Designed Characteristics and Trial Results.

	Guam	Panay	Luzon
Length between perpendiculars, ft.& inches	150-00	180-00	198-00
Beam, moulded, ft.& inches	27-00	28-00	31-00
Draft, designed, fresh water, ft. & inches	5-03	5-06	6-00
" actual, " " " " "	5-03	5-02.75	5-09.94
Displacement on designed draft, tons	387	474	615
" " actual " , tons	387	444	585
Fuel required to be carried on design draft, tons	26	23	48
" tank capacity, tons	77.48	110.08	168.39
Designed cruising radius at 10 knots	1000	1000	1200
Cruising radius on total fuel	1950*	3100	2800*
Designed speed in knots	14.5	15.0	16.0
Main Engines designed I.H.P., maximum	1950	2150	3150
" " actual " "	2048	2899	3560
Maximum speed in knots	15.6	17.7	18.3
Designed R.P.M. or propellers	320	320	310
Actual " " " at max. speed	360.3	372.4	361
" " " " " designed speed	323	290	288
Lbs. fuel per square foot H.S. at 10 knots	.17*	.259	.159
" " " " " " ddesigned speed	.578	.332	.387
Fuel oil, lbs. per I.H.P. main engines at 10 knots	3.32*	2.80	3.90*
" " " " " " designed speed	2.16	1.72	1.69

* The cruising radius of the Guam and Luzon would be considerably increased with one boiler in use instead of two. The figures on fuel oil used would also be affected.

DYSON'S METHOD OF PROPELLER ANALYSIS AND DESIGN APPLIED TO MOTOR BOATS.

By E. A. Stevens, Jr., Civil Member.

As this article is intended to be a supplement to "Screw Propellers," by Admiral C. W. Dyson, published by Simmons-Boardman Publishing Company, it is assumed that the reader is familiar with the Dyson method, therefore only the formulae and charts that do not appear in the above mentioned book will be explained in detail. All " Figures" and " Sheets" designated by numbers or numbers and one letter such as Figure 1, Sheet No. 19, and Sheet No. 20A refer to those in Dyson's book, while all figures designated by a letter as Figure A and sheets designated by a number and two letters as Sheet 20AA accompany this article.

ACCURACY OF THE METHOD.

If the model has been tested with its appendages, the pitch of the propeller accurately measured, the motor developing the same power in the boat as it did on the test block, the data on the trial accurately taken, and the propeller not working in cavitation, the calculated shaft horsepower and revolutions should be within 5 and 3 per cent respectively of the actual. When cavitation exists the same degree of accuracy cannot always be expected (especially if the wheel is heavily in cavitation), although it has been obtained in some cases.

The actual brake horsepower of the motor is determined from a curve of powers for various revolutions, data for which is obtained from shop tests, made with the throttle wide open. After the motor has been installed, the boat is run over a measured course at full power, the speed and revolutions noted and the power is obtained from the revolution-power curve of the motor. As the adjustment of the carburetor or the atmospheric conditions are not always the same in both cases, there is here a chance for error

which, in some cases, is the explanation for some of the discrepancies between the calculated and the so-called actual shaft horsepower.

It should also be noted that the brake horsepower obtained from the shop tests does not allow for the losses of the thrust bearing which should not, however, exceed 2 to 3 per cent in well-designed and lubricated thrusts. If the model has not been tested or the pitch of the propeller measured, or if the pitches of the various blades should vary, accurate results cannot be expected.

EFFECTIVE HORSEPOWER.

The problem of resistance, or effective horsepower, is too long and complicated to be dealt with here, but if accurate results are to be expected it should be obtained from calculations or, better yet, from model tests. If the speed ratio ($v \div \sqrt{L.\ W.\ L.}$) is over 1.5 it is very difficult, in most cases, to obtain the accurate effective horsepower by calculations, unless the results of a boat of about the same displacement ratio, longitudinal (prismatic) coefficient and shape is obtainable as Taylor's curves do not give the residual resistance for vessels with as high a displacement ratio as many motor boats at these speed ratios.

In many cases, models have been tested without such appendages as the shafts, struts and rudder, and where these are fitted to the boat some allowance should be made for their resistance. Sheet No. 18, Vol. II of Admiral Dyson's book gives the effective horsepower in per cent of that required for the bare hull, which gives satisfactory results for large ships but has not been found reliable for small boats. The effective horsepower required to overcome the resistance of these appendages can be calculated by the following formula:

$$e.h.p_A = .00307 \times L^3 \times v \times r_A \times c$$

e. h. p_A = Effective horsepower necessary to overcome the resistance of the appendages.

L = Length of the boat in feet.

v = Speed in knots.

r_A = Resistance in pounds of the appendages of a model 1 foot long. See Sheet No. 18 AA.

c = Constant .8 for single screw boats,
1.0 for twin screw boats,
1.2 for triple screw boats.

Before leaving the subject of effective horsepower the reader's attention should be called to a very important fact. It will be noticed from Figure A, which shows the e.h.p. curve of a 40-foot motor boat, that there is a considerable hump in the curve between the speeds of 8.5 and 17 knots, which corresponds to speed ratios of about 1.3 and 2.7. If the actual effective horsepower (shown by the solid line) is used in analyzing a propeller for a given speed, the estimated power and revolutions will be greater than the actual or if analyzing for a given power the estimated speed and revolutions will be lower. In order to correct for this the curve should be faired as shown by the dot and dash line. As there is a danger of the faired curve being too low, thereby overestimating the speed in designing a propeller, the hump should be bridged by a straight line as shown dotted. By so doing the results will be more accurate than if the actual e.h.p. curve was used without the danger of underestimating the power required for a given speed or overestimating the speed that can be obtained with the given shaft horsepower. Models of all boats, intended to run at speed ratios higher than 1.30, should be tested for speed ratios up to or exceeding 3.00 in order that this hump may be properly faired out.

After the power required by the bare hull and appendanges has been obtained, an addition of at least 10 per cent should be made in designing, to allow for the increase resistance due to the action of the water on the boat's bottom paint, head wind and rough weather.

HULL ANALYSIS.

Due to the propeller being very near the water's surface, there is a large quantity of air present, which tends to increase the slip and lower the propulsive efficiency, the latter showing up in the calculations as an increase in the power augment factor (K). For these reasons the methods shown in Admiral Dyson's book for obtaining the basic slip (S) and the power augment factor (K) are not applicable to motor boats.

Basic Slip. In all cases of both twin and single screw boats a value of .25 is taken as the basic slip.

In order to determine the power augment factor (K), it is first necessary to obtain what is known as the K Block Coefficient (K.B.C.), which is derived by correcting the actual block coefficient for the ratio of beam to length and the position of the propeller. If the screw is so placed that the lower tip is below the base line, as shown in Figures E and F, the hull will have less effect upon it than would be the case if it was placed higher. In order to correct for this in obtaining the nominal block coefficient, the draft is assumed to be equal to the draft of the propeller when this is greater than the mean draft of the hull (excluding the keel).

Letting L = Length of load water line in feet.
B = Beam at the load water line in feet.
H = Mean draft of hull excluding keel in feet.
H″ = Draft of propeller in feet.
Δ = Displacement in tons of 2240 pounds.
B.C. = Block coefficient (actual).
N.B.C. = Nominal block coefficient.

Then $\text{B.C.} = \frac{\Delta \times 35}{\text{L} \times \text{B} \times \text{H}}$ for salt water.

When the draft of the propeller (H″) is equal or less than the mean draft of the hull (H) the nominal block coefficient (N.B.C.) is equal to the block coefficient (B.C.).

When the draft of the propeller (H″) is greater than the mean draft of the hull (H).

$\text{N.B.C.} = \text{B.C.} \times \text{H} \div \text{H}''$ or

$\text{N.B.C.} = \frac{\Delta \times 35}{\text{L} \times \text{B} \times \text{H}''}$ for salt water.

Note:—The above formulas for actual and nominal block coefficients are for vessels in salt water, if in fresh water 36 should be used in place of 35.

K Block Coefficient (K.B.C.) This is obtained from Sheet No. 17B, Chart I (Vol. II of Dyson's Book) by using the nominal block coefficient (N.B.C.) as abscissa and beam divided by length (B ÷ L) as ordinate, draw a line through this point and the lower right hand corner of the chart (B.C. = 1.0 and B ÷ L = 0). The

abscissa values of the intersection of this line and lines X, Y, W, U and Z (Line U is not shown on this chart but is midway between lines W and Z), will be the values of K.B.C. as follows:

Twin Screws and wing screws of triple screw boats in all cases use line X.

Single Screws. When the so-called deadwood is carried well aft as shown in Figure D, use line Z when H″ is equal or less than H. When half of the lower blade is below the base line (H″ — H = .25D, D being the diameter of the propeller) use line U.

When the so-called deadwood is cut away as shown in Figures B and C, the shaft being carried in a strut as in Figure E or in the keel as shown in Figure F, use line W when the draft of the propeller (H″) is equal to or less than the mean draft of the hull (H).

When half of the lower propeller blade is below the base line (H″ — H = .25D) as shown in Figure F, use line Y.

When the propeller hub is on the same level as the base line, as shown in Figure E, or below the base line (H″ — H $\leqq$ .50D) use line X.

Power Augment Factor (*K*). The value of K will be equal to the square of that found from curve $C_1 - C_2$ on Sheet No. 19 (Vol. II of Admiral Dyson's book) except for values of K.B.C. less than .45. The minimum value of K being 1.10.

If radial lines are drawn through the lower right-hand corner of Sheet No. 17B, Chart I (Vol. II of Admiral Dyson's book) (K.B.C. = 1.0 and B ÷ L = 0), it will be seen that all boats plotting on the same line will have the same value of $\frac{B \div L}{1 - N.B.C.}$ and the value of K.B.C. and therefore that of K will depend upon whether line X, Y, W, U or Z is used in determining the K.B.C. With this in view Sheet No. 19AA has been prepared with $\frac{B \div L}{1 - N.B.C.}$ as abscissas and K as ordinates. The writer wishes to express his appreciation for the suggestion of this chart to Mr. H. C. E. Meyer.

For convenience the form entitled "Hull Analysis" can be used in arriving at the value of K.

FORM FOR HULL ANALYSIS.

N = Number of Propellers.
L = Length L.W.L. in feet.
B = Max. Beam at L.W.L. in feet.
H = Mean Draft of hull in feet excluding keel.
H″ = Draft of propeller (lower tip) in feet.
Δ = Displacement in tons of 2240 pounds.
B.C. = Actual Block Coeff.
$= (\Delta \times 35) \div (L \times B \times H)$ for salt water,
$= (\Delta \times 36) \div (L \times B \times H)$ for fresh water.
N.B.C. = Nominal Block Coeff. $= B.C. \times H \div H''$,
$= (\Delta \times 35) \div (L \times B \times H'')$ for salt water,
$= (\Delta \times 36) \div (L \times B \times H'')$ for fresh water.
1 — N.B.C. = One Minus Nominal Block Coefficient.
B ÷ L = Beam divided by length.
(B ÷ L) ÷ (1 — N.B.C.).
D = Diameter of propeller in feet.
(H″ — H) ÷ D.
Type of Stern: Figure B, C or D.
Use Curve X, Y, W, U or Z on Sheet No. 19AA.
K = Power Augment Factor.

PROPELLER ANALYSIS.

Basic Conditions. Sheet No. 20A (Vol. II of Dyson's book) gives various curves for the Basic (or Chart) conditions. Curves B, B′, and B″ are used in design problems only. The curves marked P.C. and $I.T_d.$ are for use in calculating propellers for ships driven by steam reciprocating engines, therefore they are not used in this work. At present the only curves that are of interest are those of Tip Speed in feet per minute (T.S.), Propulsive Coefficient ($P.C_s$) and Shaft thrust in pounds per square inch of Disk Area ($S.T_d.$).

The abscissa values of this chart represent the Projected Area Ratios of 3-bladed wheels. If a 4-bladed wheel is being analyzed use the total projected area ratio as the abscissa value for obtaining the propulsive coefficient ($P.C_s$) only. To obtain the values of tip speed (T.S.) and shaft thrust ($S.T_d$) use 3/4 of the total pro-

jected area ratio as the abscissa value. If a 2-bladed propeller is being analyzed use the total projected area ratio as abscissa in obtaining the value of $P.C_s$. and 3/2 the total projected area ratio for T. S. and $S.T_d$.

Conditions Other Than Basic. When the effective horsepower required for the given speed varies from the basic, the shaft horsepower required will not be the same as the basic. This variation depends upon the ratio of the actual to the basic effective horsepower. Letting

S.H.P. = Basic shaft horsepower.
$S.H.P_d$ = Actual shaft horsepower.
E.H.P. = Basic effective horsepower.
e.h.p. = Actual effective horsepower.
10^{Z_p} = Factor from Sheet 21AA or 21AB depending upon e.h.p. $\div$ E.H.P.

Then $S.H.P_d = S.H.P. \div 10^{Z_p}$.

(Note:—Curve No. 3 on Sheet 21 gives values for Z_p which are the logarithms of 10^{Z_p}. For the convenience of the reader Sheets 21AA and 21AB have been prepared giving the values of 10^{Z_p}).

As the hull affects the action of the propeller causing an increase in actual shaft horsepower ($S.H.P_d$) for any given e.h.p. the above formula becomes

$$S.H.P_d = \frac{S.\ H.\ P. \times K}{10^{Z_p}}$$

Letting $10^{Z_p} \div K = 10^{Z_g}$.

$S.H.P_d = S.H.P. \div 10^{Z_g}$.

Should the speed vary due, to loading or weather conditions, but the e.h.p. remaining constant, the $S.H.P_d$ will remain the same, so long as the propeller is out of cavitation. Should, however, the propeller enter cavitation due to excess thrusts the $S.H.P_d$ will raise with the lowering of the speed, e.h.p. being kept constant.

Although the subject of cavitation will be discussed more fully later, it would be well to mention here how one can ascertain if or not this condition exists. As already shown $10^{Z_g} = 10^{Z_p} \div K$. Turning to Sheet No. 21AA or 21AB, obtain the abscissa value from the curve using 10^{Z_g} as the ordinate call this $e.h.p_g \div$ E.H.P. Letting the actual speed obtained v, the basic speed V, the ratio of actual to basic speed $v \div V$. Now turn to Sheet No. 22AA

with $e.h.p_g \div E.H.P.$ as abscissa plot $v \div V$ as the ordinate. If this point falls above Curve 1 the wheel will be out of cavitation, if $v \div V$ does not exceed unity (which seldom happens). If the point falls below Curve 1 the propeller is in cavitation.

Apparent Slip and Revolutions. When cavitation does not exist the apparent slip is calculated by the following formula:

$$s = S \times \frac{S.H.P_d. \times 10^{Z_s}}{S.H.P.}$$

s = Actual or estimated apparent slip.
S = Basic apparent slip which is equal to .25.
$S.H.P_d.$ = Actual shaft horsepower.
$S.H.P.$ = Basic shaft horsepower.
10^{Z_s} = Factor from Sheet 21AC depending upon $v \div V$.
v = Actual speed in knots.
V = Basic speed in knots.

As $S.H.P. \div S.H.P_d = 10^{Z_s}$ we have

$s = S \times 10^{Z_s} \div 10^{Z_s}$.

Letting R_d = the actual revolutions per minute we have

$$R_d = \frac{101.33 \times v}{P \times (1 - s)}$$

v = Actual speed in knots as above.
P = Pitch of propeller in feet.

Analysis Forms. Having obtained the values of K from the Hull Analysis and the Basic Conditions of the propeller, we are now in a position to analyze any propeller for given conditions providing cavitation does not exist. The three forms used in analyzing are as follows:

Analysis Form No. 1. Given the speed of the boat, the effective horsepower curve and the value of K, to find the shaft horsepower and revolutions.

Analysis Form No. 2. Given the speed of the boat, the shaft horsepower and value of K, to find the effective horsepower and revolutions.

Analysis Form No. 3. Given the speed of the boat, the effective and shaft horsepowers, to find the value of K and revolutions.

FORM FOR BASIC PROPELLER CONDITIONS.

Number of Blades 2, 3 or 4.

N = Number of propellers.

D = Diameter of propeller in feet.

P = Pitch of propeller in feet.

PA ÷ DA = Projected Area Ratio of 3 Blades.
= Total Projected Area Ratio of 3-Bladed Wheels.
= 3/4 Total Projected Area Ratio of 4-Bladed Wheels.
= 3/2 Total Projected Area Ratio of 2-Bladed Wheels.

4/3 or 2/3 PA ÷ DA = Total Projected Area of 4 or 2-Bladed Wheels.

T.S. = Tip Speed in Feet per Minute (Use PA ÷ DA as abscissa on Sheet No. 20A for 2, 3 and 4-Bladed Wheels).

$$P \times R = \frac{T.S. \times P}{\pi\ D} = \text{Pitch} \times \text{Revolutions per Minute.}$$

1 — S = 1 minus Basic Slip (As S is always .25 the value of 1 — S will always be .75).

$$V = \frac{P \times R \times (1 - S)}{101.33} = \text{Basic Speed in Knots.}$$

ST_d = Shaft Thrusts in pounds per square inch of disk area (Use PA ÷ DA as abscissa on Sheet No. 22A for 2, 3 and 4-Bladed Wheels).

$$S.H.P. = \frac{N \times D^2 \times (P \times R) \times ST_d}{C} = \text{Basic Shaft Horsepower.}$$

C = 389 for 2-Bladed Wheels.
C = 291.8 for 3-Bladed Wheels.
C = 252.4 for 4-Bladed Wheels.

$P.C_s$ = Propulsive Coefficient (Use as abscissas on Sheet No. 20A the following for 2-Bladed Wheels 2/3 × PA ÷ DA, for 3-Bladed Wheels use PA ÷ DA and for 4-Bladed Wheels use 4/3 × PA ÷ DA).

E.H.P. = S.H.P. × $P.C_s$ = Basic Effective Horsepower.

ANALYSIS FORM NO. 1.

Given Speed (v), Effective Horsepower (e.h.p.) and K.
Find Shaft Horsepower ($S.H.P_d$) and Revolutions (R_d).

v = Speed in knots actual.

$v \div V$ = Speed fraction = Ratio of actual to basic speed.

e.h.p. = Effective horsepower corresponding to actual speed v.

e.h.p. $\div$ E.H.P. = Net load fraction = Ratio of actual to basic effective horsepower.

10^{Z_p} = Net load factor from Sheet 21AA or 21AB corresponding to e.h.p. $\div$ E.H.P.

$10^{Z_g} = 10^{Z_p} \div K$ = Gross load factor.

$e.h.p_g \div E.H.P.$ = Gross load factor from 21AA or 21AB corresponding to 10^{Z_g}.

$S.H.P_d$ = S.H.P. $\div 10^{Z_g}$ = Actual or estimated shaft horsepower.

10^{Z_s} = Speed factor from Sheet 21AC corresponding to $v \div V$.

$s = S \times 10^{Z_s} \div 10^{Z_g}$ = Actual or estimated apparent slip.

$1 - s$ = One minus apparent slip.

$$R_d = \frac{v \times 101.33}{P \times (1-s)} = \text{Actual or estimated R.P.M.}$$

ANALYSIS FORM NO. 2.

Given Speed (v), Shaft Horsepower ($S.H.P_d$) and K.

Find Effective Horsepower (e.h.p.) and Revolutions (R_d).

v = Speed in knots actual.

$v \div V$ = Speed fraction = Ratio of actual to basic speed.

$S.H.P_d$ = Actual shaft horsepower.

10^{Z_g} = S.H.P. $\div$ $S.H.P_d$ = Gross load factor.

$e.h.p_g \div E.H.P.$ = Gross load fraction from Sheet 21AA or 21AB corresponding to 10^{Z_g}.

$10^{Z_p} = 10^{Z_g} \times K$ = Net load factor.

e.h.p. $\div$ E.H.P. = Net load fraction from Sheet 22AA or 22AB corresponding to 10^{Z_p}.

e.h.p. = E.H.P. $\times$ (e.h.p. $\div$ E.H.P.) = Actual or estimated effective horsepower.

Actual (or estimated) apparent slip and revolutions are calculated in the same manner as in Analysis Form No. 1, beginning with 10^{Z_s}.

ANALYSIS FORM NO. 3.

Given Speed (v), Effective Horsepower (e.h.p.) and Shaft Horsepower ($S.H.P_d$).

Find K and Revolutions (R_d).

v = Speed in knots.

$v \div V$ = Speed fraction = Ratio of actual to basic speed.

e.h.p. = Effective horsepower corresponding to actual speed v.

e.h.p. $\div$ E.H.P. = Net load fraction = Ratio of actual to basic effective horsepower.

10^{Z_p} = Net load factor from Sheet 21AA or 21AB corresponding to e.h.p. $\div$ E.H.P.

$S.H.P_d$ = Actual shaft horsepower.

10^{Z_g} = S.H.P. $\div$ $S.H.P_d$ = Gross load factor.

$e.h.p_g \div$ E.H.P. = Gross load fraction from Sheet 21AA or 21AB corresponding to 10^{Z_g}.

$K = 10^{Z_p} \div 10^{Z_g}$ = Power augment factor.

Actual (or estimated) apparent slip and revolutions are calculated in the same manner as in Analysis Form No. 1, beginning with 10^{Z_g}.

CAVITATION.

As the theories or causes of cavitation are the same for motor boats as large ships, a full discussion of this subject will not be given here, as this can be obtained from "Screw Propellers," but the formulae for calculating powers and revolutions will be given in full.

Curve 1 on Sheet 22AA, accompanying this article, is taken from Sheet 22A, Chart I, Vol. II. of Dyson's book for basic slips (S) of .25. For the convenience of the reader, this sheet (No. 22AA) was prepared giving other curves, which will be explained later. When a propeller, which is being analyzed plots, using $e.h.p_g \div$ E.H.P. as abscissa and $v \div V$ as ordinate, above Curve 1 and under $v \div V = 1.0$ cavitation does not exist. When $v \div V$ exceeds unity cavitation due to excess tip speed exists, but when the wheel plots below Curve 1, it is in cavitation due to excess thrusts. As the latter kind of cavitation is the more common and the losses due to it are more serious, it will be dealt with first.

Cavitation Due to Excess Thrusts. Before an estimate of the required shaft horsepower can be made, it is necessary to obtain the load fraction at which cavitation starts with the given speed fraction $(v \div V)$. To obtain this load fraction plot on Curve 1, Sheet 22AA, a point using the value of $(v \div V)$ as the ordinate, and obtain its abscissa value, which will be called e.h.$p_1 \div$ E.H.P. Now turn to Sheet 21AA or 21AB and obtain the value of 10^{Zp} for e.h.$p_1 \div$ E.H.P. which will be called 10^{Z1}. The estimated shaft horsepower will then be:

$$S.H.P_d = \frac{S.\,H.\,P. \times 10^{Z1}}{(10^{Zg})^2}$$

Letting $(10^{Zg})^2 \div 10^{Z1} = 10^{Zc}$ we have

$$S.H.P_d = S.H.P. \div 10^{Zc}.$$

The revolutions are obtained as follows:

From Sheet 21AA or 21AB obtain the value of e.h.p. $\div$ E.H.P. for 10^{Zc} call this e.h.$p_c \div$ E.H.P.

From Sheet 22AA obtain the ordinate value of Curve 1, where its abscissa value is equal to e.h.$p_c \div$ E.H.P., which will be called $v_c \div V$. This will be the speed fraction at which cavitation starts with a gross load fraction equal to e.h.$p_c \div$ E.H.P.

Let the speed at which cavitation starts be called v_c which is equal to $V \times (v_c \div V)$.

Obtain from Sheet 21AC the value of 10^{Zs} corresponding to $v_c \div V$ which will be called 10^{Zsc}

Then $s_c = S \times 10^{Zsc} \div 10^{Zc}$

s_c = Apparent slip at speed v_c

From Figure G obtain the value of M for $v \div v_c$

$$\text{Then } R_d = \frac{101.33 \times v_c \times M}{P \times (1 - s_c)}$$

A shorter method, which however is not as accurate, is as follows:

Plot on Sheet 22AA a point using e.h.$p_g \div$ E.H.P. as abscissa and $v \div V$ as ordinate and note the value of G, which can be obtained from the curves marked $G = 1.1$, $G = 1.2$, etc. The shaft horsepower will then be:

$$S.H.P_d = S.H.P. \times G \div 10^{Zg}$$

The slip can be found by the following formula

$$s = S \times 10^{Zs} \div 10^{Zg}$$

which is the same as when cavitation is not present. The revolutions formula is also the same, namely:

$$R_d = \frac{101.33 \times v}{P \times (1-s)}$$

When the value of G is greater than 1.5 or 1.6 this method of finding the revolutions can not always be depended upon.

Cavitation Due to Excess Tip Speed. The writer has experienced only one case of this kind of cavitation and, unfortunately, the data is not very reliable. The run was made from Castle Rock at the mouth of Newport, R. I., to Race Rock, a distance of 35.4 nautical miles. It was reported that one motor was turning 1385 R.P.M., the other making only 1200, due to trouble in the circulating water system. Due to this trouble the probabilities are that the shaft horsepower was somewhat lower than that given by the revolution power curve. From the data given, the indications are, that the propellers were not in cavitation, but unfortunately the accuracy of the reported speed and revolutions are open to question. It is not at all unlikely that the actual speed through the water was less, due to favorable tides* or the revolutions were greater than reported, or both. Because of this uncertainty it would be well to keep the value of $v \div V$ below unity until more data is obtained with the higher speed fractions.

If, however, it is desired to make an estimate of the power and revolutions for wheels where the speed fraction exceeds unity, follow the method given on Pages 151 and 152, Vol. I of "Screw Propellers."

Forms for analyzing propellers when in cavitation due to excess thrusts are given in the Analysis Forms Nos. 4, 5 and 6. The first of these is for finding the shaft horsepower and revolutions when the speed, effective horsepower and value of K are given. The second is for obtaining the effective horsepower and revolutions when the speed, actual shaft horsepower and value of K are given. The third is for finding the value of K when the speed, effective horsepower and shaft horsepower are given.

* It was stated that the tide was favorable for half of the trip, and against the boat for the rest.

ANALYSIS FORM NO. 4.

(In Cavitation.)

Given Speed (v), Effective Horsepower (e.h.p.) and K.

Find Shaft Horsepower ($S.H.P_d$) and Revolutions (R_d).

v = Speed in knots actual.

$v \div V$ = Speed fraction = Ratio of actual to basic speed.

e.h.p. = Effective horsepower corresponding to actual speed (v).

e.h.p. $\div$ E.H.P. = Net load fraction = Ratio of actual to basic effective horsepower.

10^{Z_p} = Net load factor from Sheet 21AA or 21AB corresponding to e.h.p. $\div$ E.H.P.

$10^{Z_g} = 10^{Z_p} \div K$ = Gross load factor.

$e.h.p_1 \div$ E.H.P. = Abscissa value of Curve 1, Chart 22AA, where its ordinate value is $v \div V$.

10^{Z_1} = Factor from Sheet 21AA or 21AB corresponding to $e.h.p._1 \div$ E.H.P.

$10^{Z_c} = (10^{Z_g})^2 \div 10^{Z_1}$ = Cavitating load factor.

$e.h.p_c \div$ E.H.P. = Cavitating load fraction from Sheet 21AA or 21AB corresponding to 10^{Z_c}.

$S.H.P_d$ = S.H.P. $\div 10^{Z_c}$ = Actual or estimated shaft horsepower.

$v_c \div V$ = Speed fraction where cavitation begins when the load factor = $e.h.p_c \div$ E.H.P. It is found from Sheet 22AA. It is the ordinate value of Curve 1, where its abscissa value is $e.h.p_c \div$ E.H.P.

$v_c = V \times (v_c \div V)$ = Speed at which cavitation begins.

$10^{Z_{sc}}$ = Speed factor from Sheet 21AC corresponding to $v_c \div V$.

$s_c = S \times 10^{Z_{sc}} \div 10^{Z_c}$ = Apparent slip if the actual speed was equal to v_c.

$1 - s_c$

$v \div v_c$ = Ratio of actual speed to speed at which cavitation begins.

M = From Figure G for $v \div v_c$.

$$R_d = \frac{101.33 \times v_c \times M}{P \times (1 - s_c)} = \text{R.P.M.}$$

ANALYSIS FORM NO. 5.

(In Cavitation.)

Given Speed (v), Shaft Horsepower ($S.H.P_d$) and K.
Find Effective Horsepower (e.h.p.) and Revolutions (R_d).

v = Speed in knots actual.

$v \div V$ = Speed fraction = Ratio of actual to basic speed.

$S.H.P_d$ = Actual shaft horsepower.

10^{Z_c} = S.H.P. $\div$ $S.H.P_d$ = Cavitating load factor.

$e.h.p_c \div$ E.H.P. = Cavitating load fraction from Sheet 21AA or 21AB corresponding to 10^{Z_c}.

$e.h.p_1 \div$ E.H.P. = Abscissa value of Curve 1, Sheet 22AA, where its ordinate is $v \div V$.

10^{Z_1} = Factor from Sheet 21AA or 21AB corresponding to e.h.p. $\div$ E.H.P.

$10^{Z_g} = \sqrt{10^{Z_c} \times 10^{Z_1}}$ = Gross load factor if cavitation did not exist.

$10^{Z_p} = 10^{Z_g} \times K$ = Net load factor.

e.h.p. $\div$ E.H.P. = Net load fraction from Sheet 21AA or 21AB corresponding to 10^{Z_p}.

e.h.p. = E.H.P. $\times$ (e.h.p. $\div$ E.H.P.) = Effective horsepower.

The Revolutions (R_d) are found in the same way as on Analysis Form No. 4, beginning with the line $v_c \div V$.

ANALYSIS FORM NO. 6.

(In Cavitation.)

Given Speed (v), Effective Horsepower (e.h.p.) and Shaft Horsepower ($S.H.P_d$).
Find K and Revolutions (R_d).

v = Speed in knots actual.

$v \div V$ = Speed fraction.

$S.H.P_d$ = Actual shaft horsepower.

10^{Z_c} = S.H.P. $\div$ $S.H.P_d$ = Cavitating load factor.

$e.h.p_c \div$ E.H.P. = Cavitating load fraction from Sheet 21AA or 21AB corresponding to 10^{Z_c}.

$e.h.p_1 \div$ E.H.P. = Abscissa value of Curve 1, Sheet 22AA where its ordinate is $v \div V$.

10^{Z_1} = Factor from Sheet 21AA or 21AB corresponding to e.h.$p_1 \div$ E.H.P.

$10^{Z_g} = \sqrt{10^{Z_e} \times 10^{Z_i}}$ = Gross load factor if cavitation did not exist.

e.h.p. = Actual effective horsepower for speed v.

e.h.p. $\div$ E.H.P. = Net load fraction.

10^{Z_p} = Net load factor from Sheet 21AA or 21AB corresponding to e.h.p. $\div$ E.H.P.

$K = 10^{Z_p} \div 10^{Z_g}$ = Power augment factor.

The Revolutions (R_d) are found in the same way as on Analysis Form No. 4, beginning with the line $v_c \div V$.

FORMS OF PROJECTED AREA OF BLADES.

What Admiral Dyson says about the shape of the projected area holds true for motor boats as well as for ships, therefore, with the exception of the treatment of broad tip bladed wheels, the reader is referred to "Screw Propellers," pages 168 to 184, Volume I, for a full discussion of this subject.

When it is necessary to reduce the diameter from that found as necessary by the calculations for the standard shape blades, the tips of the blades should be broader. This is done by one of two ways. First:—design the wheel for a larger diameter and cut off the tips. These propellers are called "broad tip" bladed wheels. The second method is to design the wheel for a larger diameter, cut off the tips and add the area removed to the sides of the blades between the .7 radius of the basic wheel and the actual tip. Such blades are known as "fan shape"—Figures 21 and 22, page 176, Volume I of "Screw Propellers" show the "fan type" and "broad tip" blades. When designing propellers of reduced diameters "fan shape" blades should be used, as the reduction in diameter from the basic is less than that of the "broad tip" wheels and therefore the chances of errors are reduced.

When "broad tip" blades are to be analyzed, it is advisable to obtain the equivalent "fan shape" blade. This can be done as any "broad tip" blade can be considered as a "fan shape" with a smaller basic diameter although all "fan shape" blades cannot be considered as "broad tip" ones. Having the ratio of basic to

actual diameter of a "broad tip" blade this ratio for the equivalent "fan shape" type can be obtained from Figure H as shown in the following example:

EXAMPLE.

Given a "broad tip" bladed wheel as follows:

Actual diameter $= D_1 = 28''$.
Pitch $= P = 22.875''$.
Actual PA $\div$ D.A. $= .619$.
Radius of greatest 1/2 Chord (C) $= 11.375''$.
Basic diameter $= D = 2 \times 11.375 \div .70 = 32.5''$.
Ratio of basic to actual diameter $= D \div D_1 = 1.16$.

From Figure H, we find that the ratio of $D \div D_1$ of the equivalent "fan shape" is 1.055. The diameter and the projected area ratio of the basic wheel will then be as follows:

Basic diameter $= D_1 \times (D \div D_1) = 28 \times 1.055 = 29.5''$.
Basic P.A. $\div$ D.A. $=$ Actual (P.A. $\div$ D.A.) $\div (D \div D_1)^2 =$ $.619 \times 1.113 = .556$.
Pitch same as before $= 22.875$.

The propulsive efficiency of a "fan shape" wheel will be somewhat less and the revolutions higher than those of the basic wheel. The difference in propulsive coefficient is allowed for by multiplying the value of K as found from Sheet 19AA by $\sqrt[3]{D \div D_1}$. The revolutions of the "fan shape" wheel will be equal to that of the basic multiplied by $\sqrt{D \div D_1}$. In other words to analyze a "fan shape" wheel, correct K as explained above and analyze the basic wheel as if it was the one to be used with the exception of the formula for revolutions which becomes

$$R_d = \frac{101.33 \times v}{P \times (1-s)} \times \sqrt{\frac{D}{D_1}}$$

ANALYSIS PROBLEMS.

The following problems are given in order to give the reader an idea of the accuracy that may be expected when full and correct data is given, and to illustrate the use of the Analysis Forms.

The table marked " Hull Analysis Problems" shows the values of K as derived by the use of the form given for this purpose. The types vary from a high speed " V" bottom motor boat (Column C) to a Navy sailing launch (Column D), whose underwater body is quite similar to a houseboat, being rather beamy and heavy displacement.

All of the analysis problems show the use of the form for obtaining the basic conditions of the propellers.

Analysis Problem No. 1. The hull used in this problem is that shown in Column A of " Hull Analysis Problems." She was a twin-screw steamer fitted with geared turbines. Unfortunately, no torsion-meters were fitted; the power was estimated from the readings of the pressure and vacuum gauges and the revolutions. The model was tested without appendages at various displacements, the greatest being 19.2 tons while the displacement on the trial was 22.8 tons. The effective horsepower for the bare hull at trial displacement and for the appendages were estimated and are probably fairly accurate. Considering the above, together with the fact that the power developed was only about 14 per cent of the designed power, it is not surprising that there should be the differences between the $S.H.P_d$ as analyzed from the propeller and that estimated from the gauge readings.

The propellers were of the broad tip type, the characteristics of which are given in the paragraphs describing method of dealing with this type of wheel. As the ratio of the basic diameter (of the equivalent " fan shape" blade) to the actual is 1.055, the value of K to be used in this problem will be that found in Column A of " Hull Analysis Problems" multiplied by the cube root of this ratio or 1.018

$$\text{Corrected } K = 1.29 \times 1.018 = 1.312$$

Analysis Problem No. 2. The hull characteristics are those shown in Column F of " Hull Analysis Problems," and represent a type of single screw cruising motorboat of moderate speed, with the shaft exposed and carried in a strut as shown in Figure E, but were not on the model when tested. This was allowed for, however, in this problem. As the motor was not operating at full load no check can be made for the power.

Analysis Problem No. 3. This problem illustrates the method used to estimate the effective horsepower having given the shaft horsepower and the value of K. The hull represents cruising motorboat of moderate high speed and displacement ratio. The hull characteristics are given in Column B of "Hull Analysis Problems." The model was not tested, therefore no check can be made on the estimated effective horsepower.

Analysis Problem No. 4. The boat used in this problem is that in Column C of "Hull Analysis Problems," being a high speed, light displacement "V" bottom type.

Before going on to Analysis Problems Nos. 5, 6 and 7, the writer wishes to express his regrets for the lack of data in Problems Nos. 2 and 3. These Problems together with Problem No. 1 are given because of the lack of data of small boats where the propellers are not in cavitation. In the case of Problems Nos. 1 and 2, the propellers were in cavitation at the higher speed runs.

Analysis Problem No. 5. As the boat and propellers in this problem are the same as those in Problem No. 1, the remarks made there apply equally here.

Analysis Problems Nos. 6 and 7. In both of these problems the data was very complete, both being boats belonging to the U. S. Navy, that given in Problem No. 6 being a 40-foot motor boat, the other being a 36-foot motor sailing launch. The hull characteristics of the former is given in Column E, and that of the latter in Column D of "Hull Analysis Problems."

PROPELLER DESIGN.

Excepting for the hull, problems of design are the direct opposite of those of analysis. In both cases, the characteristics of the hull are (or at least should be) given, from which the value of K is obtained. In analysis problems, the characteristics of the propeller are given from which one finds if cavitation exists or not, also the required power and revolutions for a given speed. In design problems, the value of $v \div V$ and $e.h.p_g \div E.H.P.$ are assumed and the characteristics of the wheel are determined for the desired conditions. In all cases, the effective horsepower curve should be obtained if possible.

HULL ANALYSIS PROBLEMS

Column	A	B	C	D	E	F
No. Of Propellers = N	2	1	2	1	1	1
L = L.W.L.	48.75	46.32	49.97	35.375	39.33	55.3
B = Beam W.L	9.75	10.0	9.33	9.83	8.16	10.22
H = Draft Of Hull	3.333	2.09	1.66	2.20	2.16	2.19
H'' = Draft Of Propeller	3.96	3.54	2.645	2.354	3.292	4.39
Δ In Tons	22.8	10.98	7.43	10.51	6.775	13.45
B.C. = $\frac{\Delta \times 35}{L \times B \times H}$	.505	.397	.346 [X]	.481	.343	.381
N.B.C. = B.C × H ÷ H''	.425	.234	.217	.45	.225	.19
1 − N.B.C.	.575	.766	.783	.55	.775	.81
B ÷ L	.20	.216	.187	.278	.207	.185
(B ÷ L) ÷ (1 − N.B.C)	.348	.282	.239	.505	.267	.227
D = Diam Of Propeller	2.933	2.75	2.0	1.67	2.08	3.0
(H'' − H) ÷ D	†	.527	†	.092	.545	.735
Type Of Stern Fig.	B	B	B	B	B	C
Sheet 19AA. Use Curve	X	X	X	W − Y *	X	X
K	1.29	1.2	1.14	1.72	1.19	1.15

[X] Fresh water, Formula for Block Coeff. is B.C. = $\frac{\Delta \times 36}{L \times B \times H}$

† This equation does not enter into analysis of twin screw boats as Curve X Sheet 19AA is used in all cases

* As the value of (H'' − H) ÷ D is between 0 and .25 the value of K will be between those obtained from Curves W and Y

It should be borne in mind that the resistance of any vessel is considerably greater in service than under ideal conditions, that is with freshly painted bottoms, smooth sea, little or no wind and deep water and, that the effective horsepower as obtained from the tank test or from calculations are for ideal conditions. As the weather varies considerably, it is quite difficult to determine just what allowance should be made when designing propellers, but the following may be taken as representing fair averages:

Moderate and high speed cruisers 15 per cent

House boats 20 per cent

Working boats that are in use during the whole year 20 per cent

The e.h.p. curve given by the tank test or calculations should be increased by these amounts and a new curve drawn, which should be used in the calculations for design.

ANALYSIS PROBLEMS

Problem No.	1	2	3	4
S	.25	.25	.25	.25
K From Sheet 19AA	1.29	1.15	1.20	1.16
No of Blades	3	3	3	3
No of Propellers = N	2	1	1	2
Basic Diam = D	2.46	3.00	2.75	2.00
Pitch = P	1.406	3.00	2.00	2.166
PA÷DA (for 3 blades)	.556	.41	.442	.45
⅔ or ⅓ PA÷DA	—	—	—	—
T.S.	12190	8670	9350	9480
P×R=(T.S×V)÷(P×D)	3020	2760	2162	3265
1−S	.75	.75	.75	.75
V=P×R×(1−S)÷101.33	22.35	20.42	16.00	24.17
S.Td.	9.63	5.77	6.52	6.73
S.H.P.=D²×P×R=ST.÷C	12.06	491	365.5	603
C=389 for 2 bladed wheels; C=291.8 for 3 bladed wheels; C=252.4 for 4 bladed wheels				
P.Cs.	.752	.667	.644	638
E.H.P=S.H.P×P.Cs	690	327	235.4	384.5
Actual Diam = D.	2.333	3.00	2.75	2.00
$\sqrt{D \div D_1}$	1.027	1.00	1.00	1.00
$\sqrt[3]{D \div D_1}$	1.018	1.00	1.00	1.00
Corrected K from Sheet 19AA $K^{\frac{1}{3}}$ D.	1.312	1.15	1.20	1.16
v	7.45	9.00	14	20
v÷V	.3335	.441	.875	.828
e.h.p.	13.8	18.51	S.H.P$_d$ (Actual) 185	e.h.p 151.8
e.h.p.÷E.H.P	.02	.0566	10^{Z_9}=S.H.P÷S.H.P$_d$ 1.975	e.h.p÷E.H.P .395
10^{Z_P}	58.8	19.85	e.h.p$_g$÷E.H.P .52	10^{Z_P} 2.63
$10^{Z_9}=10^{Z_P}\div K$	44.8	17.26	Cavitating No	S.H.P$_d$ (Actual) 262
e.h.p$_g$÷E.H.P	.026	.065	$10^{Z_P}=10^{Z_9}\times K$ 2.37	10^{Z_9}=S.H.P÷S.H.P$_d$ = 2.30
Cavitating	No	No	e.h.p.÷E.H.P .436	e.h.p$_g$÷E.H.P .449
S.H.P$_d$=S.H.P÷10^{Z_9}	26.93	28.4	e.h.p=E.H.P×(e.h.p÷E.H.P) 103	Cavitating No
S.H.P$_d$ Actual	29.1	—	e.h.p. Actual —	K=$10^{Z_P}\div 10^{Z_9}$ 1.144
10^{Z_S}	29.2	11.95	1.99	1.77
s=S×$10^{Z_S}\div 10^{Z_9}$	.163	.173	.189	.192
1−s	.837	.827	.811	.808
$R_d=\frac{101.33\times v}{P\times(1-s)}\times\sqrt{\frac{D}{D_1}}$	406	367.5	874	1158
R_d Actual	491.6	376	875	1161

Choice of e.h.$p_g \div$ E.H.P. and $v \div V$. The choice of the gross load fraction (e.h.$p_g \div$ E.H.P.) and the speed fraction ($v \div V$) should be such, that the propeller will plot sufficiently above Curve 1, Sheet 22AA, so that cavitation due to excess thrusts will not be encountered when conditions are somewhat worse than the average, and sufficiently below $v \div V =$ unity, in order to avoid cavitation due to excess tip speed from any one or more of the following causes. First: decreased resistance due to favorable weather and for lighter draft. Second, motor operating at maxi-

ANALYSIS PROBLEMS

Problem No.	5			6		7	
S	.25			.25		.25	
K From Sheet 19 AA	1.29			1.19		1.72	
No of Blades	3			3		3	
No of Propellers = N	2			1		1	
Basic Diam = D	2.46			2.08		1.855	
Pitch = P	1.906			1.83		1.30	
P.A ÷ D.A (for 3 blades)	.556			.55		.396	
½ or ⅔ P.A ÷ D.A							
T.S	12190			12000		.8420	
P × R = (1.33 P) ÷ (V × D)	3020			3360		1878	
1 − S	.75			.75		.75	
V × P × R × (1−S) ÷ 101.33	22.35			24.86		13.89	
S.Td.	9.63			9.43		5.43	
S.H.P. = H.P × P × R × S.Td. ÷ C	12.06			470		120.3	
C = 389 for 2 bladed wheels; C = 291.8 for 3 bladed wheels; C = 252.4 for 4 bladed wheels							
P.C.s.	.572			.575		.677	
E.H.P = S.H.P × P.C.s.	690			270		81.4	
Actual Diam = D.	2.333			2.08		1.667	
$\sqrt{D \div D_1}$	1.027			1.00		1.054	
$\sqrt[3]{D \div D_1}$	1.018			1.00		1.036	
Correction A.R. from Sheet 19 AA $\sqrt{D \div D_1}$	1.312			1.19		1.781	
v	10.017	9.61	9.92		10.2		7.2
$v \div V$	.448	.43	.395		.41		.518
e.h.p.	643	491	301	S.H.P. (Actual)	97.5	S.H.P. (Actual)	30
e.h.p. ÷ E.H.P	.0932	.0716	.0436	10^{2a} = S.H.P ÷ S.H.P.	4.82	10^{2a} = S.H.P ÷ S.H.P.	4.01
10^{2P}	11.92	15.52	26.10	e.h.p.c. ÷ E.H.P.	.221	e.h.p.c. ÷ E.H.P.	.263
$10^{2g} = 10^{2P} \div K$	9.01	11.05	19.90	Cavitating	Yes	Cavitating	Yes
e.h.p$_c$ ÷ E.H.P	.122	.093	.059	e.h.p$_c$ ÷ E.H.P.	.06	e.h.p$_c$ ÷ E.H.P	.119
Cavitating	Yes	Yes	Yes	10^{2i}	19.7	10^{2i}	9.17
e.h.p$_c$ ÷ E.H.P.	.087	.079	.054	$10^{2g} = \sqrt{10^{2a} \times 10^{2i}}$	9.49	$10^{2g} = \sqrt{10^{2a} \times 10^{2i}}$	6.075
10^{2i}	12.75	14.05	20.85	$10^{2P} = 10^{2g} \times K$	11.3	e.h.p	8.5
$10^{2c} = (10^{2g})^2 \div 10^{2i}$	6.345	10.0	19.0	e.h.p$_c$ ÷ E.H.P = .0975		e.h.p ÷ E.H.P.	.1044
e.h.p.c. ÷ E.H.P	.169	.109	.057	e.h.p ÷ E.H.P = (e.h.p ÷ E.H.P)	21.83	10^{2P}	10.63
S.H.P$_c$ = S.H.P ÷ 10^{2c}	109.5	120.4	63.5	e.h.p. Actual	26	K (Corrected = $10^{2P} \div 10^{2g}$ = 1.735	1.735
S.H.P$_c$ Actual	201	119	64			K (Normal) = 1.735	1.654
$v_a \div V$	.581	.501	.406		.437		674
$v_a = V \times (v_a \div V)$	12.98	11.19	9.08		15.93		9.36
10^{2ac}	5.12	9.06	15.5		3.89		3.28
$s_c = S \times 10^{2ac} \div 10^{2c}$	.201	.200	.204		.202		2.05
$1 - s_c$	.799	.80	.796		.798		.795
$v \div v_c$	.772	.859	.971		.644		.769
M	.883	.922	.934		.817		.882
$R_d = \frac{(W \times V_c \times M) \sqrt{\frac{D}{D_1}}}{P \times (1 - s_c)}$	783	704	582		897		853
R_d Actual	799.9	693.3	579.4		919		837

mum power, when the propeller has been designed for a power somewhat less than the maximum. Third, the pitch of the propeller being less than that called for by the design. Although experience is the best guide as to the values of the gross load and speed fractions to use, the following rules will serve as a guide.

When the effective horsepower has been taken for the light draft condition, no allowance made for the increase resistance for service conditions and the maximum power of the motor is used in the calculations, $v \div V$ may be as high as .97 or .98. In this case, e.h.$p_g \div$ E.H.P. should be chosen so that $v \div V$ will plot not lower than Curve 4 on Sheet 22AA.

When the effective horsepower has been taken for average cruising draft, allowances made for the increase resistance for service conditions, and 75 to 80 per cent of the maximum power of the motor used in the calculations, $v \div V$ should not exceed .95 and should plot on or about Curve 3, Sheet 22AA.

When the boat varies considerably in draft, such as cargo carriers, the effective horsepower is taken for the maximum draft and allowances made for the increase resistance for service conditions, the value of $v \div V$ should not exceed .90. In such cases the value of e.h.$p_g \div$ E.H.P. should be such so that $v \div V$ will plot on or about Curve 2, Sheet 22AA.

If the power and/or revolutions are high, the value of e.h.$p_g \div$ E.H.P. should be taken as high as possible, as this will give a wheel with the smallest projected area ratio for the given conditions, thereby obtaining the most efficient and lightest propeller. If, however, it is found that, under these conditions the projected area ratios are too low, lower values of $v \div V$ and e.h.$p_g \div$ E.H.P. should be taken, using Curves 2, 3 or 4, as the case may be, as a guide. The basic projected area ratio of 3-blades should never be less than .20. Admiral Dyson advises .23 to .25 as the minimum.

In cases where a boat is required to tow and it is desired to have a wheel that will be efficient under such conditions, the design being made for running free, the value of $v \div V$ should be well above Curve 4. This can be done by choosing a high speed and a low power fraction.

Although the above rules do not cover all conditions they will serve, however, as a guide to the engineer in choosing the starting point in design calculations.

Design Forms. Admiral Dyson's book gives a large number of forms, any one of which may be used for designing motor boat propellers. However, the writer has found the forms used in solving the design problems given here are the most practical. The forms used in Design Problems 1, 2 and 3, together with Curves A and A′ on Sheet 20AA are fully explained in Vol. XXXIX. No. 3 of this JOURNAL. (An Additional Chart for Propeller Design by the Dyson Method.)

Design Form No. 1. Used in Design Problems Nos. 1 and 2 is used when it is desired to find the characteristics of the propeller and the shaft horsepower necessary for a given speed, effective horsepower and revolutions.

Design Form No. 2. Used in Design Problem No. 3, is used when the characteristics of the wheel is to be obtained for a given speed, revolutions and shaft horsepower. The effective horsepower delivered by the propeller can be calculated by this method. This form can also be used (as in Design Problem No. 3), when the effective horsepower curve is available, in obtaining not only the characteristics of the propeller but also the speed that may be expected when the shaft horsepower and revolutions of the motor are given. Under such conditions the speed v (Line 12, Column 1 of Design Problem No. 3) is assumed which should be as close as possible to the actual speed. The diameter, pitch and projected area ratio of the propeller and the effective horsepower delivered for the assumed speed are calculated. If the total effective horsepower delivered by all propellers (N × e.h.p. Line 26, Column 1, Design Problem No. 3) differs materially from that given by the e.h.p. curve for the assumed speed, pick from this curve the speed corresponding to the total effective horsepower as found, and start the calculations over again using the same values as before for Lines (1) to (11), unless the projected area is too low in which case lower values of $v \div V$ and $e.h.p_g \div E.H.P.$, as already explained, should be used. It is hardly ever necessary to make a third trial, as the calculated total effective horsepower on the second attempt is generally close enough for all practical cases. It will be noticed

that the diameters and projected area ratios of the two propellers found in this problem are practically the same but the pitch varies in direct proportion to the speed, therefore, when the speed from the e.h.p. curve corresponding to the total effective horsepower (N $\times$ e.h.p. Line 26) does not materially differ from the assumed speed (8 per cent or less), the diameter and projected area ratio as found in Column 1 may be used, increasing or decreasing the pitch, as the case may be, in direct ratio of the speed corresponding to the total effective horsepower to the assumed speed.

Design Form No. 3. This form, which is used in solving Design Problem No. 4, is used for obtaining the characteristic of 3-bladed propellers of reduced diameter. In such cases the diameter is limited, the actual diameter (D_1) should be taken as large as possible, the basic diameter (D) being taken at various sizes not exceeding $1.112 \times D_1$. This form can also be used in finding the minimum revolutions and character of a standard shape wheel of a given diameter by letting the basic diameter equal the actual as shown in the first column of Design Problem No. 4. After the characteristics of the propeller and the revolutions have been determined for the various wheels, curves are drawn (using either revolutions or basic projected area ratios as abscissa) from which the desired wheel may be obtained. If the shaft horsepower and revolutions of the motor are given, and it is found that the power for the assumed speed (Line 8) differs materially from that of the motor, obtain the actual propulsive coefficient. Multiply the shaft horsepower of the motor by the actual propulsive coefficient, the product being the actual effective horsepower (e.h.p.). Obtain from the effective horsepower curve the speed corresponding to the actual effective horsepower, as just found, and go through the calculations again.

Design Form No. 4. This form, which is used in solving Design Problem No. 5, is used for obtaining the revolutions and characteristics of 4-bladed wheels, and is used in the same manner as Design Form No. 5.

DESIGN PROBLEMS.

It should be noticed in all of the design problems that e.h.p. and $S.H.P_d$ refer to the effective and shaft horsepower on one shaft and N $\times$ e.h.p. and N $\times$ $S.H.P_d$ represent the total effective and

shaft horsepower. Therefore, N × e.h.p. is the effective horsepower obtained from the e.h.p. curve.

Design Problem No. 1. In this problem it is desired to find the characteristics of the propeller and the necessary shaft horsepower to drive the twin screw boat shown in Analysis Problem No. 5 (Also Problem No. 1) at 10.017 knots with the propeller turning at 800 R.P.M., which are the speed and revolutions obtained on the high speed runs of the trials. Below is a comparison of the wheels as fitted and the propellers found in this problem:

Propeller	*Analysis Problem No. 5.*	*Design Problem No. 1*
Pitch	1.906	1.505
Basic Diameter	2.46	2.367
Actual Diameter	2.33	2.367
Basic PA ÷ DA	.556	.34
Actual PA ÷ DA	.619	.34
Speed in Knots	10.017	10.017
N × e.h.p.	64.3	64.3
N × S.H.P$_d$	189.5	111.8

Thus it will be seen that, with practically the same diameter but somewhat less pitch and projected area ratio, the required power has been reduced from 189.5 (the estimated S.H.P$_d$ in Analysis Problem No. 5) to 111.8 or 41 per cent. Most of this saving in power is due to taking the propellers out of cavitation, the remainder to the reduction of the projected area ratio. At first it might seem strange that two wheels of practically the same diameter but differing as much in pitch should operate at the same revolutions, but this is explained in that the increased pitch requires greater shaft horsepower, which is, in this case, expended in increasing the revolving motion of the thrust column without delivering any additional thrust.

Design Problem No. 2. The boat used in this problem is the same as that in Analysis Problem No. 6. In this case the power has been reduced from 97.5 to 42.2 or about 56 per cent. The causes of the saving in this case are the same as in Design Problem No. 1. Such great savings, as are shown in this and the preceding problem, are only possible when the propellers are deeply in cavitation.

Design Problem No. 3. The explanation of this problem is quite lengthy, but it will be illustrated in detail, as the conditions represent those often met with in designing propellers for motor boats. The writer was called upon to design a propeller for the boat given in Column B of Hull Analysis Problems. The lines of the boat, but no e.h.p. curves were obtainable. The data given of the propeller, power of the motor, revolutions and speed obtained are as follows:

Propeller—Number of blades	3
Diameter	28″
Pitch	28″
P.A. ÷ D.A.	.425
Speed in Knots between	12.75 and 13
R.P.M.	900
B.H.P. Guaranteed	175
B.H.P. Reported as actually delivered at 900 R.P.M. on test	190

Analyses were made of the above wheel for e.h.p. using 12.75 and 13 knots for the speed and 175 and 190 for the values of $S.H.P_d$ with the following results:—

Speed	12.75	12.75	13.00	13.00
$S.H.P_d$	175	190	175	190
Estimated R_d	849	864	855	870
Estimated e.h.p.	57.4	59.7	59.2	63
Actual R_d	900	900	900	900

Estimates were then made for the effective horsepowers at various speeds and a curve drawn. This curve showed the same characteristics as that shown on Figure A. The hump on the curve was faired out (as was done on Figure A by the dot and dash line). The values of the effective horsepowers obtained from this faired curve for various speeds were as follows:—

Speed	12.75	13	13.5	14	14.5	15	15.5
e.h.p.	63.7	70.3	77.3	84	91.5	99	107

As the boat had never been run over a measured course, the speed being taken from runs between various points on Long Island Sound, and the revolutions taken from a tachometer, it is

not surprising to find the differences between the estimated and reported results. However, it was considered comparatively safe to use the values for e.h.p. from the curve as these were higher than those obtained from the propeller analysis, also to use 190 as the actual shaft horsepower. Assuming a propulsive coefficient of .50 for the new wheel, the actual effective horsepower delivered would be $190 \times .50 = 95$, which would correspond to a speed of 14.75 knots. Using this for the value of v (line 12) it was found (in the first column) that the effective horsepower (line 25 or 26) delivered is 103.5 which is considerably higher than that mentioned above. The speed, from the effective horsepower curve, for 103.5 effective horsepower is 15.3 knots, which was used as the value of v in the second column. (The values of lines 1 to 11 inclusive being the same as before), the estimated effective horsepower delivered, in this case, being 104.2 which is considered close enough to that from the curve for this speed. A stock propeller of 33 inches diameter, 24 inches pitch, and a projected area ratio of .442, with a shape practically the same as the "Dyson Standard" was ordered but, when installed, the vibrations were so great as to make it impossible to run the boat at anything like full speed. The wheel was taken off and sent to a concern making a specialty of correcting pitches and balancing propellers, where it was found to be badly out of balance and the pitch of each blade varied considerably, all of which were corrected. Runs made after the new wheel was reinstalled indicated that a speed of 14 knots was made with the throttle wide open and the engine turning 875 revolutions. An analysis for e.h.p. and R_d at 14 knots was made, which is shown in Analysis Problem No. 3. It will be noticed that the calculated and reported revolutions are very close, which tends to prove the accuracy of the data.

Before the new wheel was ordered, the owner was told that a speed of 15 knots was expected, but this speed could not be definitely predicted because of the lack of a tank test and the difficulty of estimating the effective horsepower of a boat of the speed and displacement ratios of the one in question. The difference in the estimated and actual obtained speed in this case was not due to the method of propeller design, but to the error in estimating the effective horsepower, especially for speeds over 13 knots. In

checking up some of the data, it was found that the pitch of the first wheel was never measured, the stamping on the propeller being taken as the actual pitch. If this pitch had been less, which very likely was the case, as the actual R.P.M. were higher than those obtained from the analysis, the wheel would not have been as heavily in cavitation as shown in the analysis, under which conditions the analyzed effective horsepower (e.h.p.) and revolutions (R_d) would have been higher (the S.H.P_d being the same). A comparison of the two wheels and the reported results obtained are as follows:—

Propeller	*No. 1*	*No. 2*
No. of blades	3	3
Diameter	28″	33″
Pitch	28″	24″
PA ÷ DA	.425	.442
R.P.M.	900	875
S.H.P_d	190	185
Speed in knots	13.0	14.0

The object of Design Problems Nos. 1 and 2 was to show the saving in power to be obtained by replacing propellers, that are deeply in cavitation, with suitable wheels for the same speed, effective horsepower and revolutions, while the results aimed at in Design Problem No. 3 were to obtain increased speed with the same power and revolutions.

In order to make a direct comparison the effective horsepower, speed and revolutions used in Design Problem No. 1 were the same as those in Analysis Problem No. 5, and the estimated shaft horsepower (S.H.P_d) was compared with the estimated S.H.P_d found by the analysis. The drafts of the boats used in Design Problems Nos. 1 and 2 varied considerably in service and, as the deepest draft was used in both the analysis and design problems, it was considered advisable to use .90 as the maximum value of v ÷ V so that cavitation due to excess tip speed would not be encountered at lighter drafts, and as no allowance was made for increase resistance in service, it was decided to use Curve 3, Sheet 22AA as the minimum limit. With these limits the maximum value of e.h.p_g ÷ E.H.P. is .47 which was used in both problems.

The boat in Design Problem No. 3 being a motor yacht, the draft varies very little and as maximum power of the motor was used for the propeller calculations, a value of .95 was chosen for the speed fraction, and as the accuracy of the effective horsepower curve was questionable, it was decided to keep the wheel well away from the cavitation point.

Design Problem No. 4. This problem will show the methods used to ascertain the proper revolutions, pitch and projected area ratio of a standard shape three-bladed wheel of a given diameter and the characteristics of a wheel of reduced "diameter."

The boat used is that shown in Column "F" of "Hull Analysis," which required a total e.h.p. (for hull and appendages) of 36 at 10.27 knots. In order to allow for the increased resistance in service, as already explained, an addition of 15 per cent will be made, making the e.h.p. used in the calculations 41.4. It will be assumed that 3 feet is the maximum possible diameter of the propeller.

The revolutions and the characteristics of a full diameter (Standard Shape) and various reduced diameter propellers are given in the calculations, from which curves are drawn using revolutions (R_d) as abscissas. From these curves the diameter, pitch, basic projected area ratio and actual projected area ratio (line 39) can be obtained for any desired revolutions between 462 and 518. As 10 per cent of the basic diameter ($D \div D_1 = 1.11$) is the maximum reduction, the minimum revolutions that the propeller should be operated for the above conditions is 462. Should it be desired to use lower revolutions, one or a combination of two or more of the following should be resorted to:

First—use a four-bladed wheel

Second—use a lower value of $v \div V$

Third—decrease, or eliminate, the allowance for increase resistance in service.

The disadvantage of the first is that the propulsive coefficient is reduced especially for high speed wheels. The disadvantage of the second and third is that the propellers will enter cavitation due to excess thrusts sooner but, to offset this the propulsive coefficient will be slightly increased as long as cavitation does not exist.

Design Problem No. 5. This problem is the same as the preceding except for the following: First: the hull is of the "House Boat" type being very beamy and heavy displacement and fitted with twin screws. Second: four-bladed propellers are used.

Wherever possible the use of "reduced diameter" propellers should be avoided, as the reduction in revolutions is not accompanied with increase propulsive coefficient unless the full diameter can be used while the weight of the motors, shafting and propellers will be increased with the lower revolutions.

If the following facts are kept in mind when working on design problems considerable labor may be saved.

First: all propellers plotting on Curve 2, Sheet 22AA, will have practically the same diameter and pitch providing the speed of the boat, shaft horsepower and revolutions are constant. The projected area ratio will increase and the propulsive coefficient decrease with the lower values of $e.h.p_g \div E.H.P.$ and $v \div V$. The same can be said of Curves 3 and 4.

Second: propellers plotting on Curve 4, Sheet 22AA, will have larger diameters and projected area ratios and less pitch and propulsive efficiency than those plotting on Curves 3 or 2, $S.H.P_d$ and R_d being the same in all cases.

Third: for any given value of $e.h.p_g \div E.H.P.$ the maximum propulsive coefficient will be obtained when $v \div V$ plots on Curve 1. As $v \div V$ increases the projected area ratio increases and the pitch and propulsive efficiency decreases. If the propeller plots below Curve 1 (in cavitation due to increase thrusts) the projected area ratio and propulsive efficiency decreases and pitch increases as the value of $v \div V$ decreases.

Fourth: for any given value of $v \div V$ the diameter and projected area ratio will increase while the pitch decreases as the value of $e.h.p_g \div E.H.P.$ decreases. The propulsive coefficient decreases if the propeller plots above (out of cavitation) and increases if below (in cavitation) Curve 1, Sheet 22AA, as the $e.h.p_g \div E.H.P.$ decreases, $v \div V$ being constant.

DESIGN PROBLEMS

Problem No.		1	2		3	
Number of Blades		3	3		3	
Number of Propellers = N		2	1		1	
(1)	K	1.29	1.19		1.20	
(2)	S	.25	.25		.25	
(3)	1-S	.75	.75		.75	
(4)	v ÷ V	.90	.90		.95	
(5)	e.h.p ÷ E.H.P	.47	.47		.50	
(6)	10^{x9}	2.19	2.19		2.055	
(7)	$10^{x9} = 10^{x6} \times K \times (4) \times (1)$	2.825	2.605		2.47	
(8)	e.h.p ÷ E.H.P.	.368	.398		4.19	
(9)	10^{x8}	1.375	1.375		1.175	
(10)	$s = S \times 10^{x8} \div 10^{x9} = (2) \times (9) \div (6)$	.157	.157		.149	
(11)	1-s	.843	.843		.851	
(12)	v	10.017	10.2		14.75	15.3
(13)	Rd	800	918		900	900
(14)	$P = \frac{V \times 101.33}{R_d \times ((12)^3)} = \frac{(12) \times 101.33}{(13) \times (11)}$	1.505	1.335		1.938	2.01
(15)	$R = \frac{R_d}{V \div V} \times \frac{1-s}{1-S} = \frac{(13) \times (11)}{(4) \times (3)}$	999	1147		1082	1082
(16)	N x e.h.p	64.3	26.33	N x S.H.P$_d$	190	190
(17)	N	2	1	N	1	1
(18)	$e.h.p = \frac{N \times e.h.p.}{N} = \frac{(16)}{17}$	32.15	26.33	$S.H.P_d = \frac{N \times S.H.P_d}{N} = \frac{(16)}{(17)}$	190	190
(19)	$E.H.P = \frac{e.h.p.}{e.h.p \div E.H.P} = \frac{(18)}{(8)}$	87.4	66.2	$S.H.P = S.H.P_d \times 10^{x9} = (18) \times (6)$	390	390
(20)	$A = \frac{E.H.P \times R \times C'}{10,000 \times P} = \frac{(19) \times (15) \times C'}{10,000 \times (14)}$	16.93	16.6	$A' = \frac{S.H.P \times R \times C'}{10,000 \times P} = \frac{(19) \times (15) \times C'}{10,000 \times (14)}$	63.6	61.2
C' = 348 for 2 bladed wheels, = 2.918 for 3 bladed wheels, = 2.824 for 4 bladed wheels						
(21)	P.A ÷ D.A. From Sheet 20 A.A.	.34	.337	P.A ÷ D.A. From Sheet 20 A.A	.454	.45
(22)	4/3 or 2/3 P.A ÷ D.A = 2/3 or 1/3 x (21)	——	——	4/3 or 2/3 P.A ÷ D.A.	——	——
(23)	P.C$_s$.	.714	.716	P.C$_s$.	.634	.638
(24)	$S.H.P = E.H.P \div P.C_s = (19) \div (23)$	122.5	92.4	E.H.P ÷ S.H.P x P.C$_s$ = (19) x (23)	247.2	248.8
(25)	$S.H.P_d = S.H.P \div 10^{x9} = (24) \div (6)$	55.9	42.2	$e.h.p = E.H.P \times \frac{e.h.p}{E.H.P} = (24) \times (8)$	103.5	104.2
(26)	N x S.H.P$_d$ = (17) x (25)	111.8	42.2	N x e.h.p = (17) x (25)	103.5	104.2
(27)	T.S.	7430	7390	T.S	9560	9480
(28)	$D = \frac{T.S.}{\pi \times R} = \frac{(27)}{\pi \times (15)}$	2.367	2.05	$D = \frac{T.S}{\pi \times R} = \frac{(27)}{\pi \times (15)}$	2.795	2.788

DESIGN PROBLEM No 4

	No. Of Propellers = N = 1		No Of Blades			3
(1)	D_1 = Actual Diam.	3·0				
(2)	$D \div D_1$	1·0	1·033	1·067	1·11	
(3)	$\sqrt{D \div D_1}$	1·0	1·065	1·033	1·054	
(4)	$\sqrt[3]{D \div D_1}$	1·0	1·011	1·022	1·036	
(5)	K_1 (K from Sheet No 19AA)	1·15				
(6)	K (Corrected) = $K_1 \times \sqrt{D \div D_1}$ = (5) × (4)	1·15	1·163	1·176	1·192	
(7)	S	·25				
(8)	V	10·27				
(9)	N × e.h.p. Total e.h.p. on all Shafts	41·4				
(10)	N = Number of Shafts	1				
(11)	e.h.p. = N × e.h.p ÷ N = (9) ÷ (10)	41.4				
(12)	e.h.p ÷ E.H.P.	·63				
(13)	$10^{x \cdot 3}$	1·617				
(14)	$10^{x \cdot 3} P = 10^{x \cdot 3} \times K$ = (13) × (6)	1·86	1·88	1·902	1·928	
(15)	e.h.p ÷ 4.P.	·551	·545	·539	·532	
(16)	V ÷ V	·95				
(17)	E.H.P = e.h.p. ÷ (e.h.p ÷ E.H.P) = (11) ÷ (15)	75·1	76	76·8	77·8	
(18)	V = v ÷ (v÷V) = (8) ÷ (16)	10·81				
(19)	D = Basic Dia. = $D_1 \times (D \div D_1)$ = (1) × (2)	3·00	3·10	3·20	3·33	
(20)	D^2	9·00	9·61	10·24	11·09	
(21)	$ET_4 = (E.H.P \times C) \div (D^2 \times V) = [(17) \times C] \div [(20) \times (18)]$	2·225	2·107	1·998	1·870	
(22)	C = 3·84 for 2 bladed wheels ; C = 2·88 for 3 bladed wheels					
(23)	P.A ÷ D.A. = Proj. Area Ratio of 3 blades	·225	·216	·208	·203	
(24)	⅔ PA ÷ DA = Total Proj. A. Ratio of 2 bladed wheels	—	—	—	—	
(25)	P_2 or $P C_2$	·77	·772	·774	·775	
(26)	[illegible] or S.H.P. = E.H.P ÷ $P C_2$ = (17) ÷ (25)	97.6	98.4	99.2	100.4	
(27)	[illegible] or S.H.P. ÷ [illegible] S.H.P ÷ $10^{x \cdot 3}$ = (26) ÷ (13)	60.4	60.9	61.4	62.1	
(28)	1 − S = 1 − (7)	·75				
(29)	B. from Sheet 20A	·0547	·058	·0595	·0604	
(30)	B × D = (28) × (19)	·17	·1798	·1904	·2011	
(31)	V ÷ (1−S) = (18) ÷ (28)	14·42				
(32)	$T_4 = (B \times D \times V) \div (1-S)$ = (30) × (31)	2·452	2·593	2744	2·90	
(33)	$10^{x \cdot 8}$	1·175				
(34)	$s = S \times 10^{x \cdot 8} \div 10^{x \cdot 3}$ = (7) × (33) ÷ (13)	·182				
(35)	(1−s)	·818				
(36)	$R_d = \frac{101.33 \times P}{P \times (1-s)} \times \sqrt{\frac{D}{D_1}} = \frac{101.33 \times (6) \times (3)}{(32) \times (35)}$	518	498	478	462	
(37)	[illegible] or N × S.H.P. = (10) × (27)	60.4	60.9	61.4	62.1	
(38)	$TA_2 \div DA_1 = TA \div DA \times (D \div D_1)^2$ = (23) × (2)²	·225	·2305	·2367	·250	
(39)	$\frac{2}{3} TA_2 \div DA_1 = \frac{2}{3} TA \div DA \times (D \div D_1)^2$ = (24) × (2)²					

DESIGN PROBLEM No 5

No Of Propellers = N = 2 — No Of Blades = 4

(1)	D_1 = Actual Diam	3·0				
(2)	$D \div D_1$	1·0	1·033	1·067	1·11	
(3)	$\sqrt{D \div D_1}$	1·0	1·0165	1·033	1·054	
(4)	$\sqrt[3]{D \div D_1}$	1·0	1·011	1·027	1·036	
(5)	K_1 = K from Sheet No 19 AA	1·55				
(6)	K (Corrected) = $K_1 \times \sqrt[3]{D \div D_1}$ = (5) × (4)	1·55	1·567	1·584	1·606	
(7)	S	·25				
(8)	v	8·0				
(9)	N × e.h.p. Total e.h.p. on all Shafts	29				
(10)	N = Number of Shafts	2				
(11)	e.h.p. = N × e.h.p ÷ N = (9) ÷ (10)	14·5				
(12)	e.h.p$_s$ ÷ E.H.P.	·30				
(13)	$10^{x \cdot s}$	3·49				
(14)	$10^{x \cdot s} P_s = 10^{x \cdot s} \times K$ = (13) × (6)	5·41	5·47	5·53	5·61	
(15)	e.h.p ÷ E.H.P.	·1975	·195	·193	·191	
(16)	$v \div V$	·78				
(17)	E.H.P. = e.h.p ÷ (e.h.p ÷ E.H.P) = (11) ÷ (15)	73·4	74·4	75·1	75·9	
(18)	$V = v \div (v \div V)$ = (8) ÷ (16)	10·26				
(19)	D = Basic Dia. = $D_1 \times (D \div D_1)$ = (1) × (2)	3·0	3·10	3·20	3·33	
(20)	D^2	9·0	9·61	10·24	11·09	
(21)	1 − S = 1 − (7)	·75				
(22)	$P_s R = (V \times 101·33) \div (1-S)$ = (18) × 101·33 ÷ (21)	1387				
(23)	$P_s R \times D^2$ = (22) × (20)	12480	13330	14200	15370	
(24)	E.H.P × 252·41 = (17) × 252·41	18530	18780	18950	19150	
(25)	B' = E.H.P × 252·41 ÷ $P_s R \times D^2$ = (24) ÷ (23)	1·485	1·409	1·335	1·246	
(26)	PA ÷ DA = Basic Prof. A. Ratio of 3 bladed	·223	·216	·208	·20	
(27)	⅓ PA ÷ DA = Total Basic PA Ratio of 4 bladed wheels	·297	·288	·277	·267	
(28)	Exp. or $P.C_a$	·74	·745	·75	·753	
(29)	[illegible] S.H.P. = E.H.P ÷ $P.C_a$ = (17) ÷ (28)	99·2	99·8	100·1	100·8	
(30)	[illegible] S.H.P$_s$ = S.H.P ÷ $10^{x \cdot s}$ = (29) ÷ (13)	28·43	28·6	28·7	28·90	
(31)	B From Sheet 20 A	·0568	·0578	·0591	·0605	
(32)	B × D = (31) × (19)	·1705	·1792	·1891	·2015	
(33)	V ÷ (1 − S) = (18) ÷ (21)	13·68				
(34)	$P = (B \times D \times v) \div (1-S)$ = (32) × (33)	2·333	2·451	2·588	2·758	
(35)	$10^{x \cdot s}$	2·12				
(36)	$s = S \times 10^{x \cdot s} \div 10^{x \cdot s}$ = (7) × (35) ÷ (13)	·152				
(37)	1 − s = [illegible]	·848				
(38)	$N_d = \frac{10 \cdot 3 \times v}{P_s (1-s)} \times \sqrt{\frac{D}{D_1}} = \frac{101·33 \times (8)}{(34)} \times \sqrt{(3)}$	410	396	382	365	
(39)	[illegible] N × S.H.P$_s$ = (10) × (30)	56·8	57·2	57·5	57·6	
(40)	$PA_1 \div DA = PA \div DA \times (D \div D$ [illegible] (26) × (2)^2	·223	·229	·237	·246	
(41)	⅔ $PA_1 \div DA_1 = \frac{2}{3}$ [illegible] $DA \times (D \div D)^2$ [illegible] (27) × (2)^2	·297	·305	·316	·328	

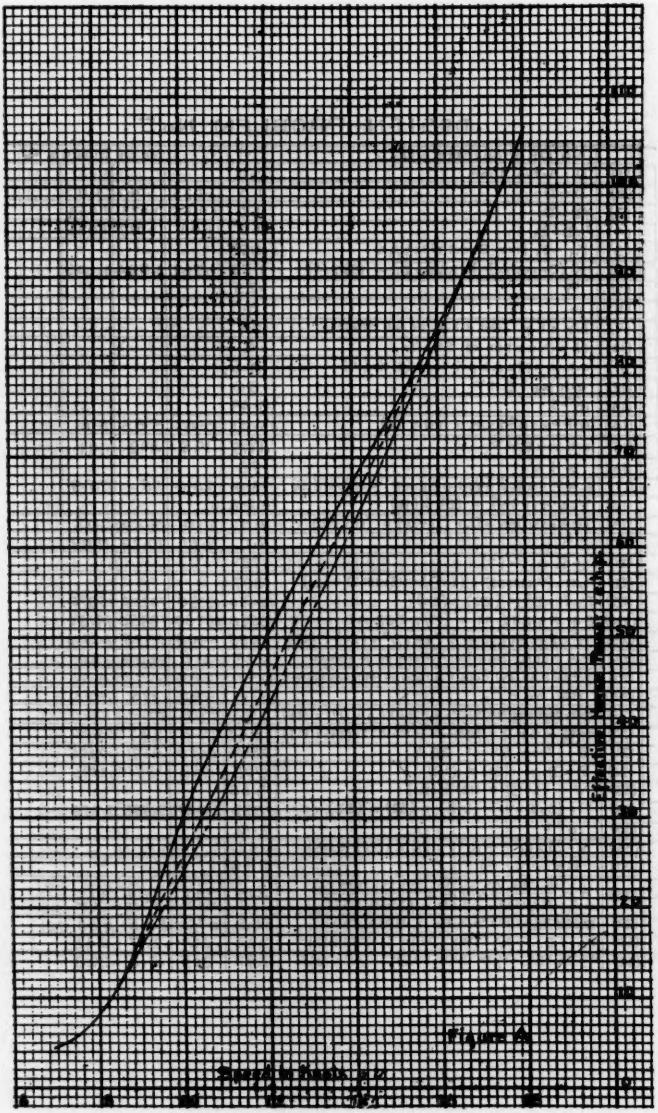


Figure 4.

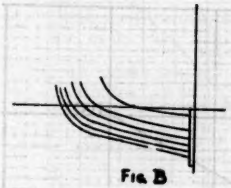


FIG. B

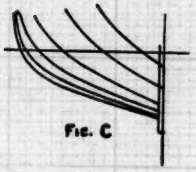


FIG. C

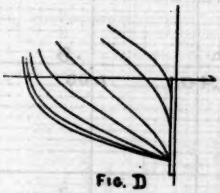


FIG. D

FIG. E

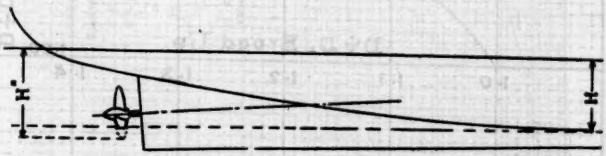
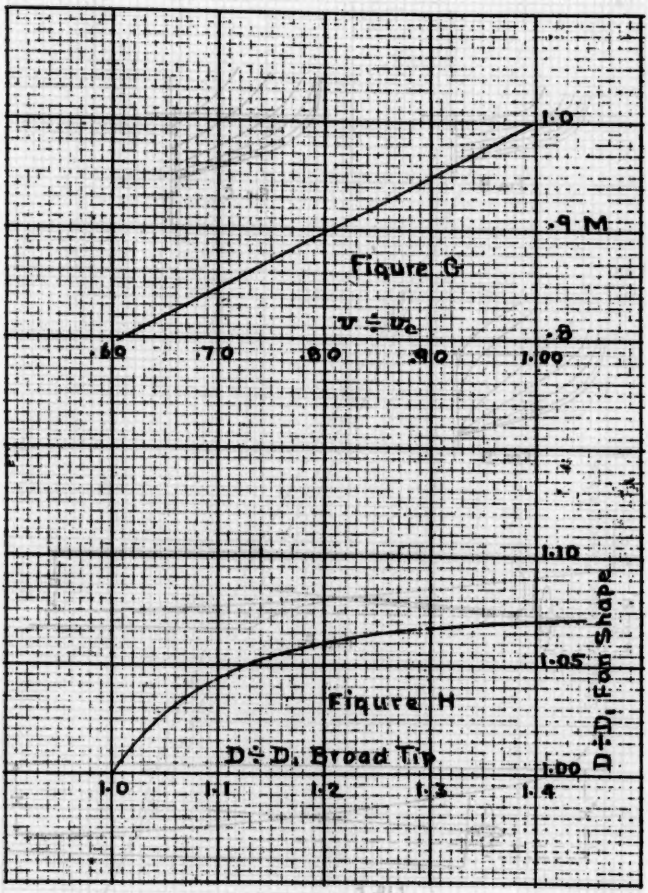
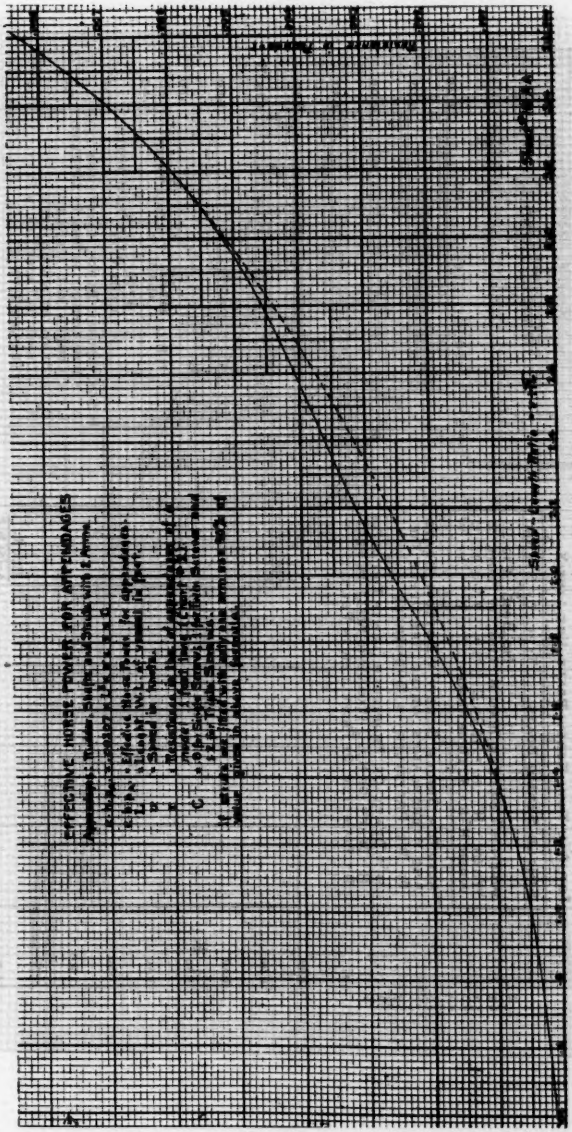
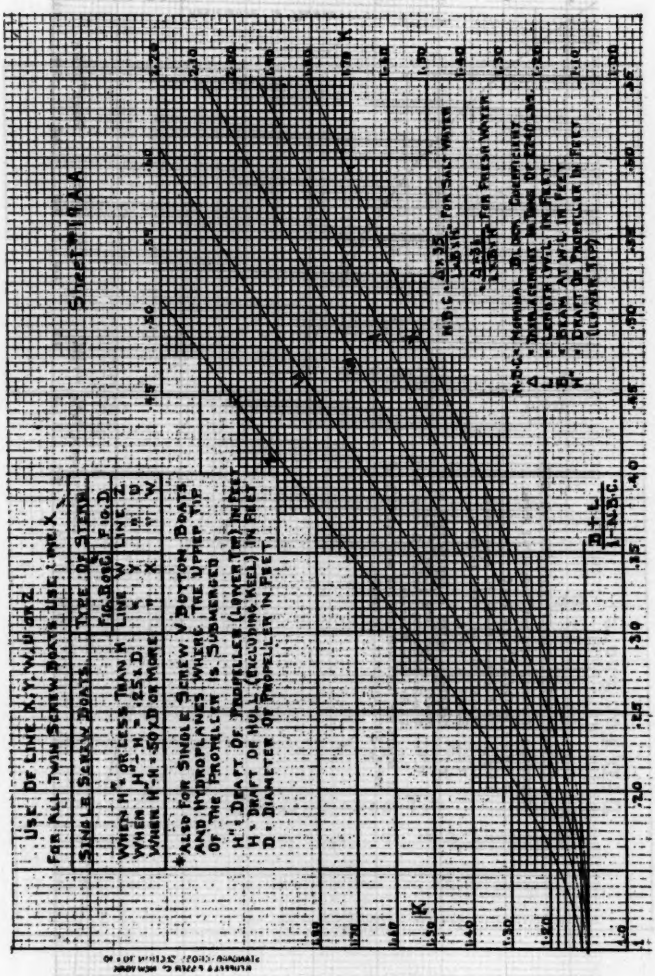
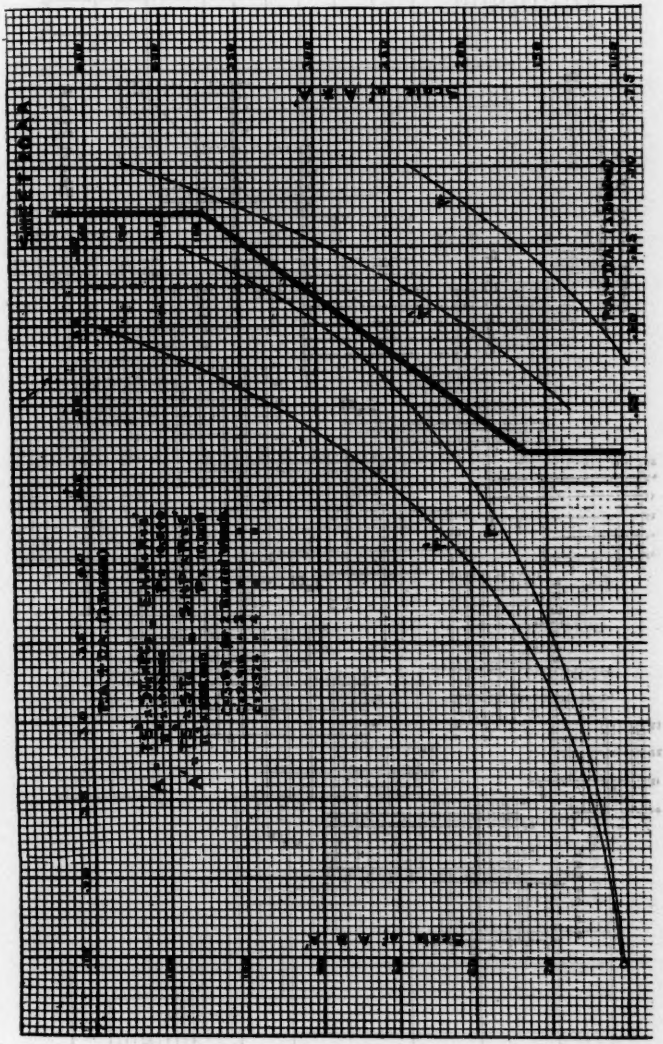


FIG. F


1.0
.9 M
Figure G
V ÷ Vc
.8
.60
.70
.80
.90
1.00
1.10
1.05
1.00
D÷D₁ Shape
Figure H
D÷D₁ Broad Tip
1.0
1.1
1.2
1.3
1.4




USE OF LINE X, Y, W, OR Z.
FOR ALL TWIN SCREW BOATS, USE LINE X.
SINGLE SCREW BOATS
TYPE OF STERN
FIG. 8 OR 9
FIG. 10
WHEN H = OR LESS THAN K
USE LINE W
USE LINE Z
WHEN H = K + .25 D
USE LINE Y
USE LINE D
WHEN H = K + .50 D OR MORE
USE LINE X
USE LINE W
* ALSO FOR SINGLE SCREW V BOTTOM BOATS AND HYDROPLANES WHERE THE UPPER TIP OF THE PROPELLER IS SUBMERGED
H = DEPT. OF PROPELLER (LOWER TIP) IN FEET
H = DRAFT OF HULL (INCLUDING KEEL) IN FEET
D = DIAMETER OF PROPELLER IN FEET.
SHEET #19 A
K
B+L / 1-N.B.C.
N.B.C. = .0035 FOR SALT WATER
.0031 FOR FRESH WATER
N.B.C. = NOMINAL BLADE COEFFICIENT
B = BREADTH OF BLADE IN FEET
L = LENGTH OF BLADE IN FEET
D = DRAFT OF PROPELLER IN FEET
(LOWER TIP)



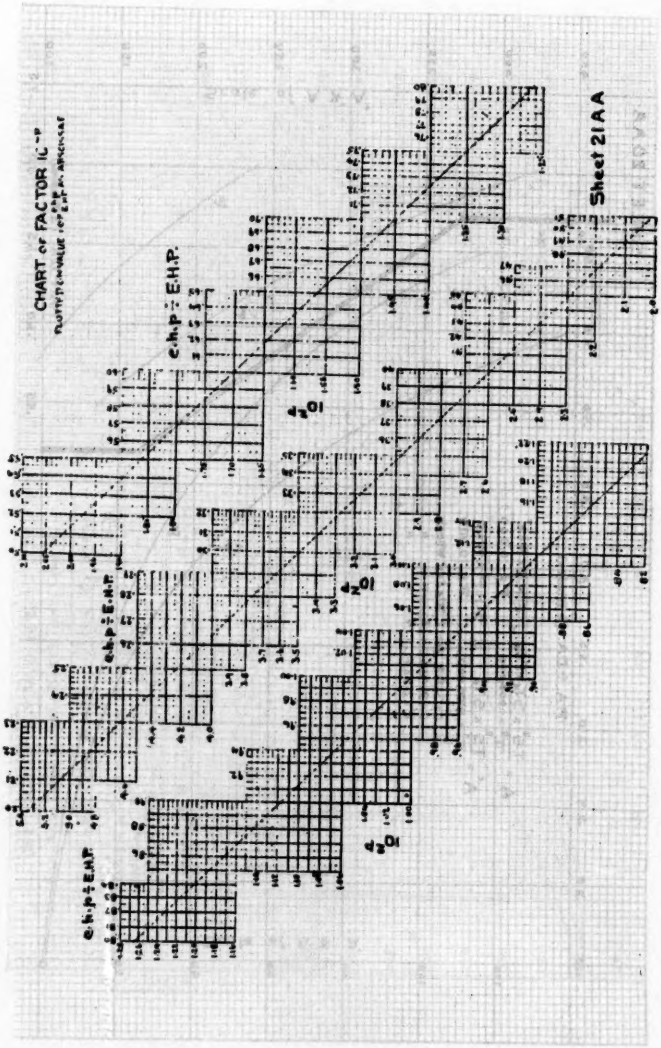
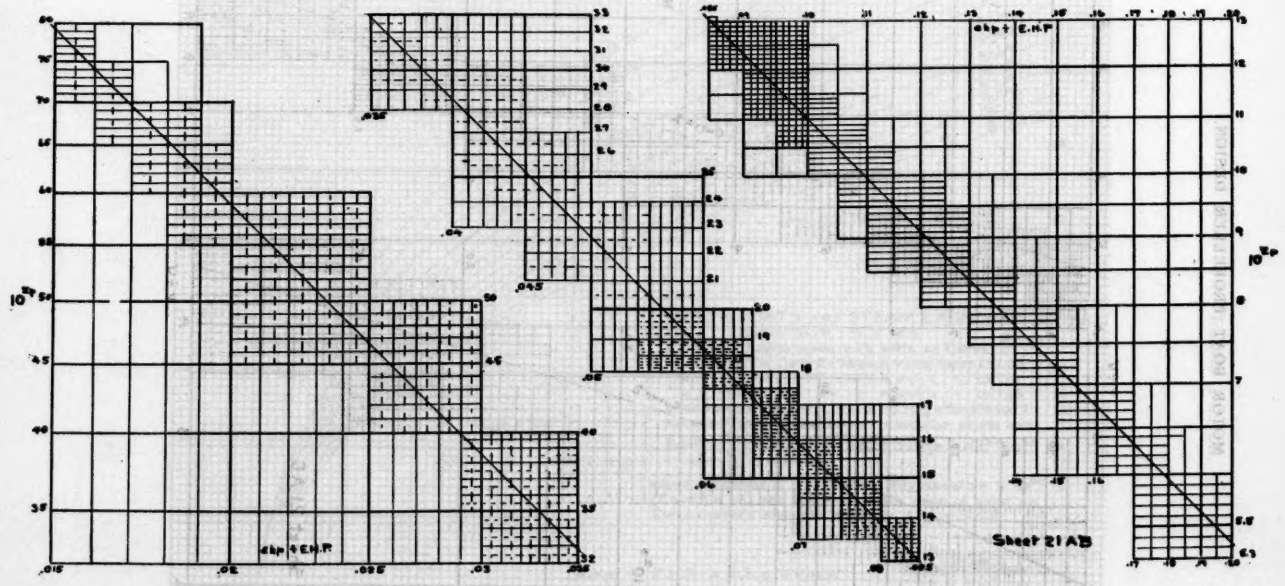
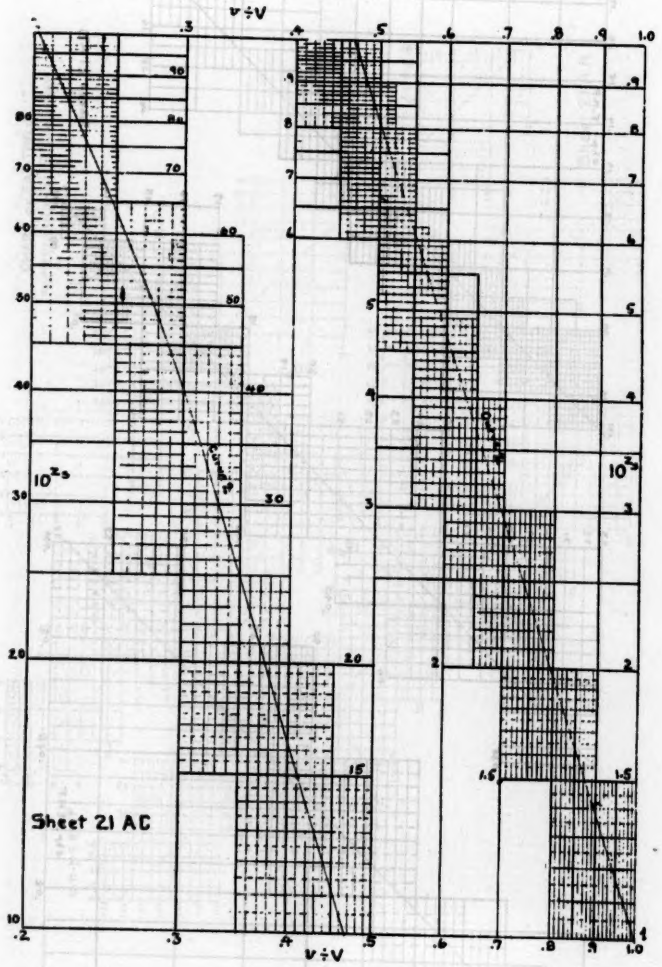
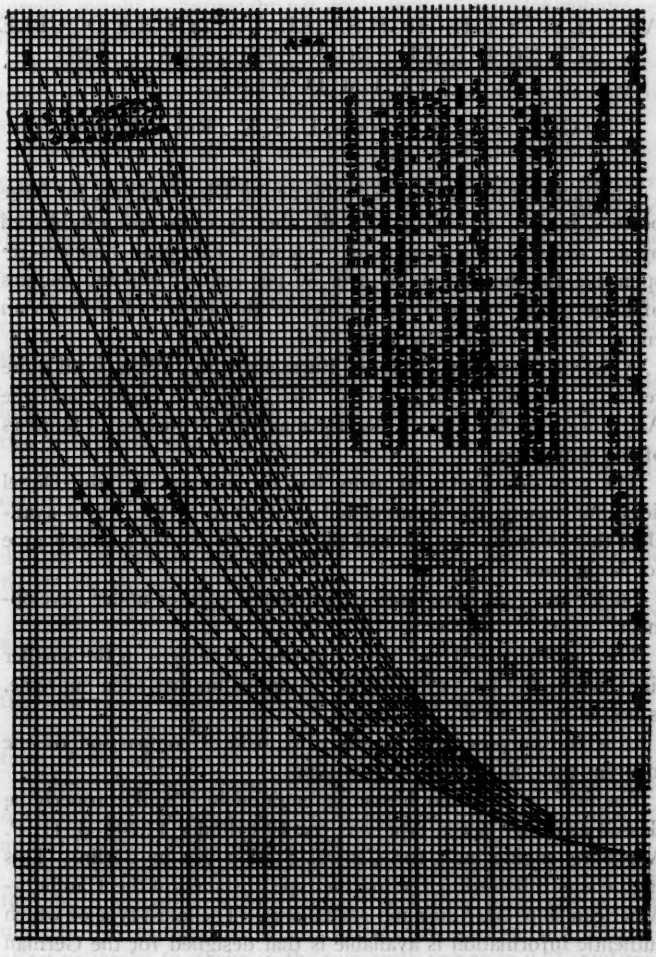


CHART OF FACTOR IC-P
Sheet 21 A A
e.h.p. : E.H.P.


Sheet 21 AB


v÷V
.3
.4
.5
.6
.7
.8
.9
1.0
10^{Z_3}
10^{Z_3}
Sheet 21 AC
v÷V



A STUDY OF THE HIGH SPEED, TWO CYCLE DOUBLE ACTING DIESEL ENGINE, FOR SMALL AND MEDIUM POWERS.

By CAPTAIN A. M. PROCTER, U. S. N. (RET.), MEMBER.

The fact that the two cycle double acting Diesel engine, for large powers, is destined to supplant all other types is now established by an overwhelming weight of evidence. European builders responsible for the designs of over 80 per cent of existing engines of large power are building engines of this type or have such an engine in the development stage.

A list of these firms includes Burmeister and Wain, until quite recently the chief protagonists of the four cycle engine; and the Augsburg branch of M. A. N., for many years successful builders of the same type.

The great A. E. G. company has recently entered the Diesel field, and having the accumulated experience of the world available, this company has adopted the two cycle double acting engine for large powers.

Sulzer and Fiat have large experimental engines under development.

The fact that this type offers superior advantages in the smaller sizes, where the four cycle trunk piston engine has been supreme, is not as yet generally recognized.

It is the purpose of this study to make a comparison of the inherent possibilities of the two types.

The first two cycle double acting engine which could properly be classed as a high speed type was the 10 cylinder 11,700 B. H. P. Augsburg engine built for Central Station use in Berlin. This engine was followed by the marine type, installed in the Hamburg-American ships *St. Louis* and *Milwaukee*. The latest type of which authentic information is available is that designed for the German warship *Ersatz Preussen*. Details of this engine are not available at present, but there is little doubt that it is a development of the previous designs.

Designed for a service where there is a premium on low weight it is certain that the piston speed has been increased, and that the specific weight,—the weight per cubic inch of cylinder volume,—has been decreased.

In order to establish a basis for an estimate of the design factors involved a comparison of two types of Augsburg engines is shown on Figure 1.

The two engines shown have the same power per cylinder and represent about the same stage in the development of the two types. The four cycle engine has been drawn from the published drawings of the 10 cylinder 3000 B. H. P. German submarine engine designed at Augsburg during the war.

The two cycle engine shown on Figure 1 is the Berlin engine reduced to a stroke of 15 inches and a bore of 12 inches. The piston speed and M.E.P. are the same as for the Berlin engine.

It is quite certain that the rating of the four cycle engine is higher than would be possible under the conditions for which the two cycle engine was designed.

When installed in the Groba power station, after the war this engine, in 10 cylinder units, was rated at 1700 B.H.P.

It is also certain that the conditions for the two cycle engine did not require the same reduction in specific weight as for the four cycle submarine engine.

In view of the different conditions the following preliminary assumptions are justified:

First:—That the two cycle, double acting engine can be run at the same piston speed as the four cycle trunk piston engine.

Second:—That the mean indicated pressures will be proportional to the ratios of the effective strokes to full stroke.

Third:—That the specific weight of the two cycle engine, without scavenging blowers, will be 50 per cent greater than that of a solid injection trunk piston engine.

The comparison shown on Figure 2 has been made by applying the above assumptions to a more recent four cycle engine. In making this comparison it has been assumed that the ratio of effective stroke to stroke in the two cycle engine is 0.75, and that the mechanical efficiencies are 0.90 for the two cycle and 0.85 for the four cycle engine.

This comparison has been extended to cover all cylinder sizes up to 30.0 inches in diameter. The results are shown on Figure 3. For a power per cylinder of 390 B.H.P., which is the maximum rating of a two cycle, double acting, engine with a 12 inch cylinder diameter, the results are as follows:—

	Two cycle D. A.		Four cycle trunk piston	
			V Type	Line
Ratio stroke-bore	1:1	1.25:1	1:1	1:1
Cylinder diameter, inches	12.0	12.0	20.9	20.9
Stroke, inches	12.0	15.0	20.9	20.9
Wt. cu. in. cyl. vol., pounds	3 0	3.0	1.7	2.0
Wt. B. H. P. engine alone, pounds	9.9	12.0	31.0	36.0
Wt. B. H. P. scavenging blowers, pounds	1.0	1.2	....	....
Wt. B. H. P. total, pounds	10.9	13.2	31.0	36.0
Ratios	1.0 :	1.21 :	2.84 :	3.30
		1.0 :	2.35 :	2.73

From the above comparison the conclusion is inevitable that, applying the greatest possible factor of caution, the two cycle double acting engine, on the weight per B.H.P. basis, is something more than twice as good as the four cycle trunk piston engine.

As a check on the assumptions made a point for the engines of the *Ersatz Preussen* is shown on Figure 3.

The authorized statement in regard to these engines gives the total power as 50,000 B.H.P. and the weight per B.H.P. for the engine alone as 8 kilos (17.6 pounds).

Information from unofficial sources is to the effect that the power is divided between four engines driving two shafts through reduction gears.

In view of the marked preference which the Augsburg designers have shown for the 10 cylinder engine, the probable number of cylinders is forty, which would give 1250 B.H.P. per cylinder.

With this figure and the known weight of 17.6 per B.H.P., the resultant engine characteristics check absolutely with the assumptions upon which the curves on Figure 3 are based.

The probable characteristics of the *Ersatz Preussen* engines are as follows:—

Stroke	550 mm. 21.6 in.
Cylinder diameter	550 mm. 21.6 in.
M. E. P. (Brake)	76.0 pounds square inch
Piston speed	1578 feet per minute
R. P. M.	438
Specific weight, engine alone	3.0 lbs. cu. in. cyl. vol.

An installation for a submarine drive, using two cycle double acting engines, is shown on Figure 4. In this plan four 6 cylinder engines are mounted on one common bedplate.

The two after engines drive the propellers through reduction gears.

The two forward engines drive through geared generators and motors. The generators can be used for battery charging and the motors for submerged running. All engines are nonreversible, and one half power is available for backing with the electric drive.

Weights of existing submarine engines are not available for comparison.

With the published weights of two 10 cylinder Augsburg engines,—414,000 pounds,—it is certain that with clutches, slow speed motors, and connecting shafting an installation for 6000 shaft horsepower would weigh at least 600,000 pounds, an increase of 45 per cent over the weights of the two cycle installation, for 27 per cent less power. The weight per S.H.P. would be 100 pounds, as compared with 57 pounds shown on Figure 4.

The engine there shown is moderately rated at 1500 foot piston speed. At 1600 feet per minute the shaft horsepower would be increased to about 7700 S.H.P.

The arrangement shown on Figure 4 gives great flexibility, and has obvious advantages as a solution of the special problems of submarine drive.

A glance at this plan, where but one half of the power is in the electric drive, is sufficient, however, to show the heavy price paid for this form of drive as compared with the straight geared drive. With a gear efficiency of 98 per cent the shaft horsepower would be over 8000, the weight would be about 180,000 pounds and the weight per S.H.P. would be about 22.5 pounds for the propulsive machinery and the auxiliaries necessary to its operation.

The estimates which have been made in this study are based on known data of existing engines. There is an abundance of evidence available that lower specific weights, higher piston speeds, and higher mean indicated pressures are possible.

It should be possible in the near future to build two cycle double acting engines, developing 400 B.H.P. per cylinder, on a weight of less than 10 pounds per B.H.P., including all necessary auxiliaries.

In larger units the weight per B.H.P. will increase in the ratio of the linear dimension or in the ratio of the square root of the powers, up to powers of about 1200 B.H.P. per cylinder.

Above that the weight will increase at a slightly higher rate owing to the lower piston speeds possible in large engines.

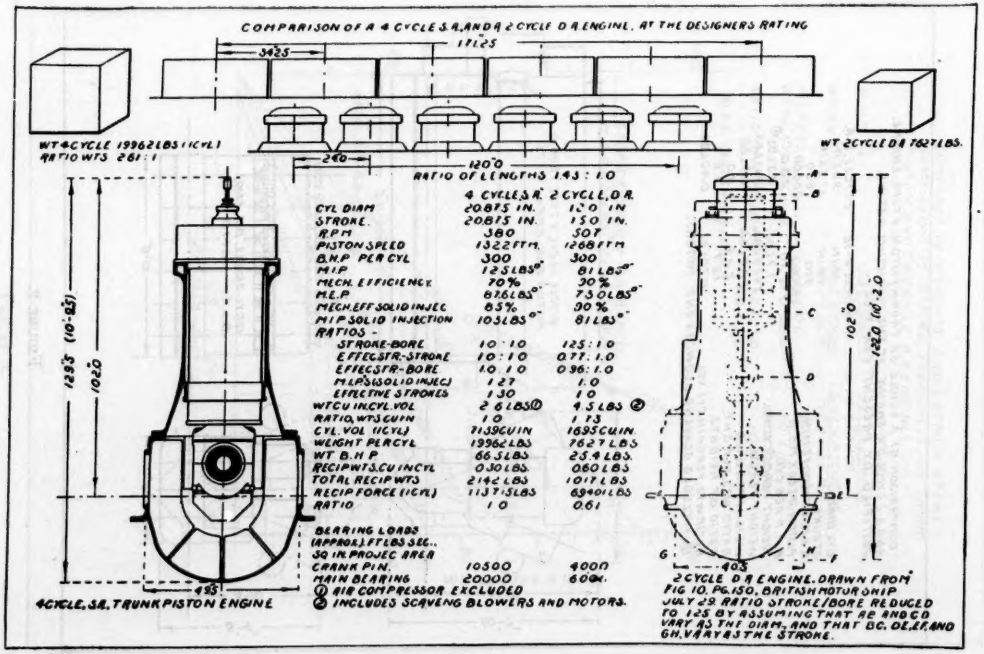

COMPARISON OF A 4 CYCLE S.A. AND A 2 CYCLE D.A. ENGINE, AT THE DESIGNERS RATING
17'-2.5"
3'-4.25"
WT 4 CYCLE 19362 LBS (1 CYL)
RATIO WTS 2.61 : 1
WT 2 CYCLE D.A. 7627 LBS.
2'-4.0"
12'-0.0"
RATIO OF LENGTHS 1.43 : 1.0
4 CYCLE S.A.
2 CYCLE, D.A.
CYL DIAM 20.875 IN. 12.0 IN
STROKE. 20.875 IN. 15.0 IN.
R.P.M. 380 307
PISTON SPEED 1322 FT/M 1268 FT/M
B.H.P. PER CYL 300 300
M.I.P. 125 LBS/IN² 81 LBS/IN²
MECH. EFFICIENCY 70% 90%
M.E.P. 87.6 LBS/IN² 73.0 LBS/IN²
MECH. EFF SOLID INJEC 85% 90%
M.I.P. SOLID INJECTION 103 LBS/IN² 81 LBS/IN²
RATIOS -
STROKE-BORE 1.0 : 1.0 1.25 : 1.0
EFFEC. STR.-STROKE 1.0 : 1.0 0.77 : 1.0
EFFEC. STR.-BORE 1.0 : 1.0 0.96 : 1.0
M.I.P.'S (SOLID INJEC.) 1.27 1.0
EFFECTIVE STROKES 1.30 1.0
WT CU IN CYL. VOL 2.6 LBS ① 4.5 LBS ②
RATIO, WTS CU IN 1.0 1.73
CYL. VOL (1 CYL) 7139 CU IN 1695 CU IN
WEIGHT PER CYL 19362 LBS 7627 LBS
WT B.H.P. 66.5 LBS 25.4 LBS.
RECIP WTS. CU IN CYL 0.30 LBS. 0.80 LBS.
TOTAL RECIP WTS 2142 LBS 1017 LBS.
RECIP FORCE (1 CYL) 11371 LBS 6940 LBS.
RATIO 1.0 0.61
BEARING LOADS (APPROX.) FT LBS SEC.. SQ IN. PROJEC AREA
CRANK PIN. 10500 4000
MAIN BEARING 20000 6000
① AIR COMPRESSOR EXCLUDED
② INCLUDES SCAVENG BLOWERS AND MOTORS.
12'-9.5" (10'-8.5")
10'-2.0"
4'-9.5"
4 CYCLE, S.A. TRUNK PISTON ENGINE
A
B
C
D
E
F
G
H
10'-2.0"
12'-0" (10'-2.0")
4'-0.5"
2 CYCLE D.A. ENGINE. DRAWN FROM FIG 10, PG. 150, BRITISH MOTOR SHIP JULY 29 RATIO STROKE/BORE REDUCED TO 1.25 BY ASSUMING THAT AB AND CD VARY AS THE DIAM., AND THAT BC, DE, EF AND GH, VARY AS THE STROKE.

FIGURE 1.

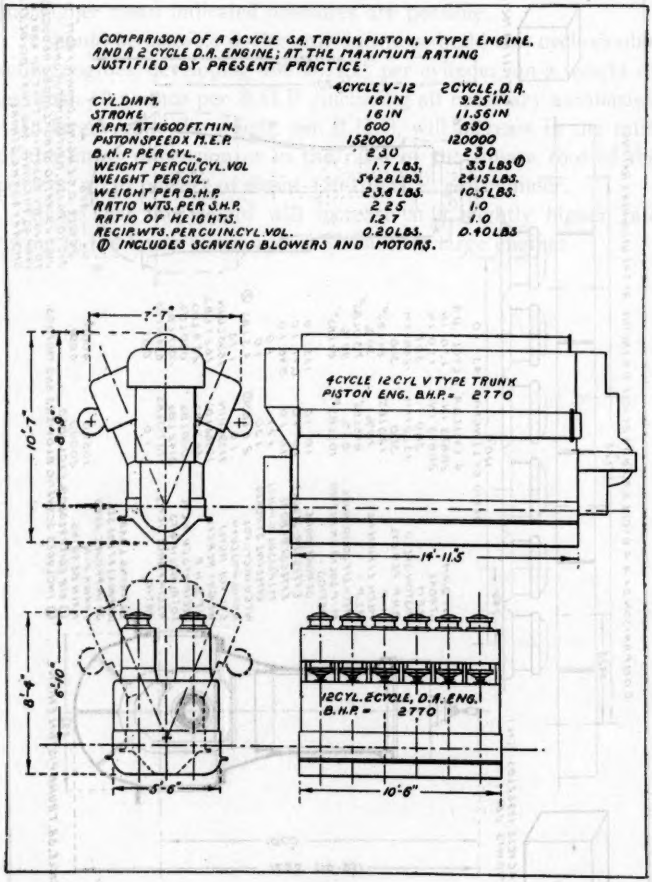

COMPARISON OF A 4 CYCLE S.A. TRUNK PISTON, V TYPE ENGINE, AND A 2 CYCLE D.A. ENGINE; AT THE MAXIMUM RATING JUSTIFIED BY PRESENT PRACTICE.
4 CYCLE V-12
2 CYCLE, D.A.
CYL. DIAM. 16 IN 9.25 IN
STROKE 16 IN 11.56 IN
R.P.M. AT 1600 FT. MIN. 600 830
PISTON SPEED X M.E.P. 152000 120000
B.H.P. PER CYL. 230 230
WEIGHT PER CU. CYL. VOL 1.7 LBS. 3.3 LBS (1)
WEIGHT PER CYL. 5428 LBS. 2415 LBS.
WEIGHT PER B.H.P. 23.6 LBS. 10.5 LBS.
RATIO WTS. PER S.H.P. 2.25 1.0
RATIO OF HEIGHTS. 1.27 1.0
RECIP. WTS. PER CU. IN. CYL. VOL. 0.20 LBS. 0.40 LBS
(1) INCLUDES SCAVENG BLOWERS AND MOTORS.
7'-7"
10'-1"
8'-9"
4 CYCLE 12 CYL V TYPE TRUNK PISTON ENG. B.H.P. = 2770
14'-11.5
8'-4"
6'-10"
5'-6"
12 CYL. 2 CYCLE, D.A. ENG. B.H.P. = 2770
10'-6"

FIGURE 2.

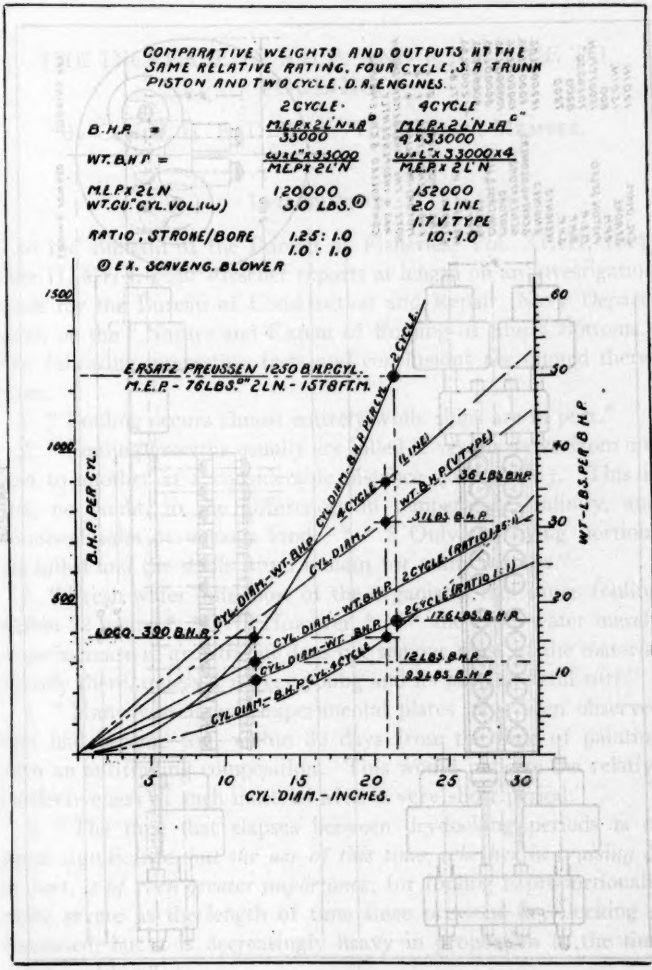

COMPARATIVE WEIGHTS AND OUTPUTS AT THE SAME RELATIVE RATING, FOUR CYCLE, S.A. TRUNK PISTON AND TWO CYCLE, D.A. ENGINES.
2 CYCLE.
4 CYCLE
B.H.P. =
WT. B.H.P. =
M.E.P. x 2L'N
WT. CU. "CYL. VOL. (W)
3.0 LBS. ①
20 LINE
1.7 V TYPE
RATIO, STROKE/BORE
1.25 : 1.0
1.0 : 1.0
10 : 1.0
① EX. SCAVENG. BLOWER
ERSATZ PREUSSEN 1250 B.H.P. CYL.
M.E.P. - 76 LBS. 2L.N. - 1578 FT.M.
2 CYCLE
4 CYCLE
(I LINE)
WT. B.H.P. (V TYPE) .36 LBS B.H.P.
.31 LBS B.H.P.
CYL. DIAM.
CYL. DIAM. - WT. B.H.P.
2 CYCLE (RATIO 1.25 : 1)
2 CYCLE (RATIO 1 : 1)
.176 LBS. B.H.P.
.12 LBS. B.H.P.
.092 LBS. B.H.P.
LOCO. 330 B.H.P.
CYL. DIAM. - B.H.P. CYL. 4 CYCLE
B.H.P. PER CYL.
WT.-LBS. PER B.H.P.
CYL. DIAM. - INCHES.

FIGURE 3.

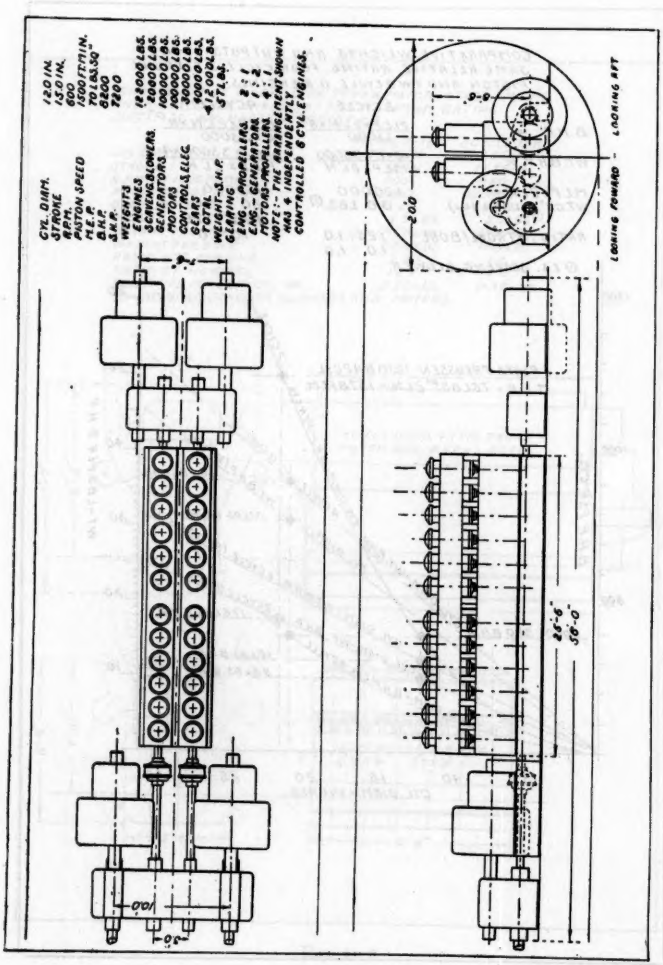

CYL. DIAM. 120 IN.
STROKE 120 IN.
R.P.M. 600
PISTON SPEED 1200 FT/MIN.
M.E.P. 70 LBS./SQ.IN.
I.H.P. 8200
S.H.P. 7200
WEIGHTS
ENGINES 1820000 LBS.
SCAVENGE BLOWERS. 80000 LBS.
GENERATORS 100000 LBS.
MOTORS 100000 LBS.
CONTROLS & ELEC. 20000 LBS.
GEARS 10000 LBS.
TOTAL 4120000 LBS.
WEIGHT-S.H.P. 571 LBS.
GEARING
ENG.-PROPELLERS. 2 : 1
ENG.-GENERATORS. 4 : 1
MOTORS-PROPELLERS. 4 : 1
NOTE:- THE ARRANGEMENT SHOWN HAS 4 INDEPENDENTLY CONTROLLED 8 CYL. ENGINES.
LOOKING FORWARD
LOOKING AFT
25'-6"
58'-0"

FIGURE 4.

THE INCREASE IN S.H.P. AND R.P.M. DUE TO FOULING.

By COMDR. H. F. D. DAVIS, U. S. NAVY, MEMBER.

INTRODUCTION.

In the Bulletin of the Bureau of Fisheries, Vol. XLIII, 1927, Part II, Mr. J. Paul Visscher reports at length on an investigation made for the Bureau of Construction and Repair, Navy Department, on the "Nature and Extent of Fouling of Ships' Bottoms." The following interesting facts and conclusions are quoted therefrom:

1. "Fouling occurs almost entirely while ships are in port."

2. "Fouling growths usually are killed if vessels move from one port to another at a considerable distance (500 miles). This is due, no doubt, to the differences in temperature, salinity, and dissolved salts of various kinds. * * * Only the living portions are killed and the shells often remain for many months."

3. "Fresh water kills most of the organisms that cause fouling within 72 hours * * *." "However * * * the fresh water merely stops increase in growth but does not remove most of the material already there, unless it is very young and its parts are still soft."

4. "Many vessels and experimental plates have been observed that had become foul within 30 days from the time of painting with an antifouling composition. This would indicate the relative ineffectiveness of such material after a very short period."

5. "The time that elapses between drydocking periods is of great significance, *but the use of this time, whether in cruising or in port, is of even greater importance,* for fouling is proportionally more severe as the length of time since previous dry docking is increased, but it is decreasingly heavy in proportion to the time spent cruising."

6. "Certain species of barnacles grow at a very rapid rate, attaining a size of 2 inches and becoming sexually mature within 60 days." "After a free-swimming period of from three days to

several weeks, the cyprids (larvæ form of barnacles) attach to some substratum and metamorphose into the adult tpye of barnacle."

7. "Fouling can be predicted from a knowledge of seasonal abundance of larval organisms in given ports."

8. As might naturally be expected with animal and vegetable growths, there is a seasonal periodicity which varies with the climate and doubtless other factors. " * * * the periods of active fouling vary with the kinds of fouling. Thus, the hydroids and algæ are late winter and early spring forms; while many of the barnacles, the oyster, and the bryozoon Bugula are late spring and summer forms, and some barnacles and the turnicate molgula are late summer or early autumn forms. Each of these is found earlier in southern waters, as at Beaufort and Norfolk, than in the cooler and more northern waters, as at New York and specially Boston." It was found that the growths are inactive in the localities mentioned during October, November and December.

Captain Henry Williams (CC), U. S. N., in an article entitled "Notes on Fouling of Ships' Bottoms and the Effect on Fuel Consumption" published in the A. S. N. E. JOURNAL of May, 1923, shows that appreciable increase in resistance has been noted without noticeable fouling due to sea growths, and suggests that such increase may be due to a change in the paint film or to the formation of an impalpable slime.

It is obvious that the fouling of a ship's bottom changes the skin friction, a measure of which change is given by the increase in the horsepower necessary to maintain a given speed. For, as indicated in Taylor's "Speed and Power of Ships," pps. 61-63, the change in resistance in pounds may be expressed thus: $R = f SV^n$ from which it follows that the change in horsepower may be expressed thus: $\frac{RV}{326} = \frac{f SV^{n+1}}{326}$ in which f is a coefficient, n an exponent, S is the wetted surface in square feet, and V is the speed in knots. In order to collect data on the increase in power necessary to maintain given speeds as fouling progresses, foul bottom standardization trials must be run following the usual clean bottom trials.

Such standardization trials of a battleship and of a destroyer were carried out at different times during periods approximating a year in each case. It is the purpose of this paper to investigate the results of these trials in the light of the above quoted facts and conclusions.

METHOD OF PRESENTING DATA.

Rules have been published to the effect that the increase in horsepower due to fouling is uniformly from 1 to 2 per cent per month.

Taking a clue from the facts and conclusions quoted, it is evident that no effort should be made to show a regular increase in horsepower as time goes on, but rather that variable changes in horsepower for the same speed should be looked for and the reasons for such changes sought.

Accordingly the original standardization data of the two ships having been plotted in curves of S. H. P. *vs.* Speed and Speed *vs.* R. P. M., the increases in S. H. P. (shaft horsepower) and R. P. M. (revolutions per minute) corresponding to speeds 10, 12, 15, 18 and 19 knots for the battleship, and to speeds 10, 15, 20, 25 and 29 for the destroyer were determined for each standardization; these increases were plotted through the year; and the histories of the ships, *i.e.*, the ports visited and the lengths and speeds of voyages were determined.

The results of this work are shown plotted in Figures 1 and 2 and the cruising data for the ships are given in Tables I and II.

DISCUSSION OF THE FIGURES AND DATA.

The foul bottom standardization trials having been run by the ships' officers without the special assistance usually given for such trials, the data leave something to be desired at a number of points. It will be noted, however, that, for the destroyer particularly, the data generally and also for the trials of each standardization, are remarkably consistent. Data for the battleship for the standardization of April 19th for speeds above 14 knots were not obtained, due to a casualty. This was particularly unfortunate in that the effects on the fouling of the long cruise from Panama to San Pedro would have been shown by these data.

In plotting, the spots of each standardization for corresponding speeds have been connected to one another with a view to identifying them in this way. It is not intended to indicate thereby that the rate of increase of resistance was necessarily uniform during the periods elapsing between such spots.

FIGURE 1, FOR THE DESTROYER.

It will be noted that upon being undocked at Boston, a clean bottom standardization was run at Rockland on 3 October; the first foul bottom trial was run on 9 November, the second on 13 December and the third on 22 March; the vessel operated in New England waters until 31 March when she sailed for Guantanamo. As might be expected there was an increase in horsepower (about one-third of the maximum observed) during the first two months. In this connection it may be noted that the vessel did relatively little steaming. From 13 December to 22 March there was practically no increase, indicating no growth of fouling plants or animals, which result should properly be expected in cold waters at that season of the year.

The standardization at Guantanamo on 12 April, one week after arrival shows a falling off of horsepower, indicating that the change from cold to warm water had killed the fouling animals and plants, that the cruise of 1400 miles had washed off some of the putrefied material, and that the stay of one week in a tropical port had not been long enough to allow the local organisms to grow to a very great extent.

In connection with this trial it may be noted that the R. P. M. point for the 29 knot curve is out of line with the corresponding S. H. P. point and all other data for that day. The original data not being available to the writer this marked discrepancy has been left as found.

At the next standardization, at Guantanamo on 10 May, the tropical growths had reached a point where the increase in horsepower was greater than it had been in March. The ship left Cuban waters on 10 May, proceeded to Boston for 4 days, then operated in the vicinity of Hampton Roads until 11 June, when she returned to Boston for a 13-day stay. On 26 June she was standardized at Rockland. It is evident that, although the change

from tropical Cuban waters to the colder northern waters may have killed the organisms acquired in the south, the stay of a month in northern waters during the most favorable growing period was sufficient to bring the resistance to a still higher point than was observed on 10 May.

During the remainder of the summer the vessel remained in New England waters, the successive standardization trials of 24 July and 11 September showing still greater increases in horsepower as should have been expected.

On 16 September the ship proceeded to Hampton Roads and operated in that vicinity until 29 October when she returned to Boston for ten days. From 13-19 November the last series of trials were run at Rockland. The results of these final trials indicate very definitely that some of the growth of the summer had been killed by the voyages south and back, or by the coming of cold weather, or by all influences combined, and that the resistance had fallen to a point where it was probably due largely to the remaining calcareous parts of the barnacles and other sea animals.

FIGURE 2, FOR THE BATTLESHIP.

Due to the missing data the story here is not so clear. Following undocking on 31 October, the ship remained at Bremerton until 17 November when she sailed for San Pedro. The standardization runs on 23-24 November and 4-8 January, giving rather erratic results particularly at low powers, definitely show small increases in horsepower, indicating, as should be expected, that the growth of marine animals and plants is slight if any at that time of the year. The standardization of 6 February confirmed these findings.

The vessel on 8 February sailed for Panama, operated in southern waters until 31 March, and arrived back at San Pedro on 11 April. There were no standardization trials during this period to tell the story of fouling.

Unfortunately, also, the trials of 19 April were interrupted and give little on which to hazard a guess at conditions. One would naturally expect a rather rapid growth while the ship was in southern waters, which growth would be killed and reduced in effect by the 2800-mile trip back to northern waters.

The standardization of 17-18 May shows an increase in resistance as would be expected at this time of the year, the vessel having operated but little and entirely in the vicinity of San Pedro for the month following 19 April.

For the seven weeks preceding the standardization in August the ship operated in Puget Sound waters making only short runs. Under these conditions and at that time of the year, a rapid growth of animal and plant organisms would naturally be expected, confirmed by the results of the last trials of this ship, 22-23 August.

SOME CONCLUSIONS.

The form taken by the spots and data suggests the possibility of determining in quantitative form, at least within reasonable limits, (*a*) the rate of growth of fouling during periods which are unfavorable to the growth of marine animals and plants, (*b*) the rate of growth of fouling when conditions are favorable to the growth of such organisms, and (*c*) the effect on fouling of a known amount of a change in conditions incident to a voyage from one place to another. With such measures and after careful consideration of the history of a given ship, it should be possible to give at least an "engineer's guess" as to the effect of fouling on the performance of such ship. It is clear that any similar trials made in the future should be so timed as to obtain the data required to compute (*a*), (*b*) and (*c*) rather than set at monthly or any other regular intervals.

The influence of fouling of ships' bottoms on performance is so great that definite information thereon should be available to all officers having to do with the logistic problems of the Fleet. Therefore further carefully planned and conducted trials should be run.

Docking periods for economy should be varied according to the service of the different ships.

TABLE No. 1

CRUISING REPORT - DESTROYER

Port Visited	Arrived (Date)	Departed (Date)	Time Underway		Distance Steamed (Knots)
			Hours	Minutes	
Boston		Oct. 1			
Rockland	Oct. 2	" 3	16	00	149
"	" 3	Nov. 1	8	20	136
Boston	Nov. 2	" 5	14	35	150
Rockland	" 6	" 8	16	12	149
"	" 8	" 8	0	40	12
"	" 9	" 9	0	58	14
"	" 9	" 10	8	54	131
Portland	" 10	" 13	6	15	72
"	" 12	" 19	5	13	75
Boston	" 19	" 25	7	45	147
Rockland	" 26	" 30	11	27	149
Boston	" 30	Dec. 13	8	08	149
Rockland	Dec. 13	" 18	8	48	151
Boston	" 18	Mar. 20	6	55	149
Rockland	Mar. 20	" 22	9	17	149
Boston	" 23	" 27	15	56	149
Provincetown	" 27	" 28	4	48	48
Newport	" 28	" 31	8	44	122
Guantanamo	Apr. 5	Apr. 12	129	52	1426
"	" 12	" 14	9	03	144
"	" 14	" 16	14	50	146
"	" 16	" 18	9	43	124
"	" 18	" 21	11	05	41
"	" 21	" 24	11	10	95
"	" 24	" 25	8	03	114
"	" 25	" 28	7	40	95
"	" 28	" 30	12	27	169
"	" 30	May 1	6	25	85
"	May 1	" 10	7	33	82
Boston	" 14	" 18	100	35	1660
Southern Drill Grs.	" 20	" 21	49	38	670
"	" 21	" 22	5	32	76
"	" 22	" 23	4	53	104
Hampton Roads	" 23	" 26	13	06	145
Southern Drill	" 26	" 27	11	28	187
" Grounds.	" 27	" 28	14	59	235
Hampton Roads	" 28	June 2	8	45	125
South'n Dr.Grnds.	June 2	" 3	9	46	140
"	" 3	" 4	9	45	168
Hampton Roads	" 4	" 5	11	45	161
Sou.Drill Grounds	" 5	" 6	7	33	114
Hempton Roads	" 6	" 9	11	10	135

TABLE No. 1.

CRUISING REPORT - DESTROYER
(Continued)

Port Visited	Arrived (Date)	Departed (Date)	Time Underway Hours	Minutes	Distance Steamed (Knots)
Southern Drill G.	June 9	June 10	11	50	136
Boston	" 11	" 24	27	12	515
Rockland	" 24	" 26	8	06	151
"	" 26	July 3	8	30	175
Bath, Me.	July 3	" 5	3	46	69
Rockland	" 5	" 11	3	58	69
Boston	" 12	" 14	15	15	151
Rockland	" 14	" 24	8	26	151
Boston	" 25	" 25	22	39	316
"	" 25	" 28	10	16	180
Newport, R.I.	" 30	Aug. 1	55	-	875
Melville, R.I.	Aug. 1	" 1	1	15	5
New York City	" 1	" 2	10	58	170
Newport	" 2	" 3	8	45	165
Rockland	" 4	" 9	20	15	246
Newport	" 10	" 11	21	14	246
"	" 18	" 19	158	27	2290.6
Melville	" 19	" 19	-	52	5
Newport	" 19	" 26	1	59	40
Boston	" 27	" 27	20	25	300
Rockland	" 28	Sept. 11	9	15	151
"	Sept. 11	" 12	10	35	156
Boston	" 12	" 15	11	28	237
Newport	" 15	" 16	9	59	170
Hampton Roads	" 17	" 18	27	40	408
Sou. Drill Grounds	" 18	" 20	4	0	54
Hampton Roads	" 20	" 22	20	30	305
Sou. Drill Grounds	" 22	" 26	2	50	35
Hampton Roads	" 26	Oct. 3	34	22	372.5
Portsmouth, Va.	Oct. 3	" 7	2	34	8.0
Sou. Drill Grounds	" 7	" 11	5	02	20.7
Hampton Roads	" 11	" 13	3	10	49.3
Lynnhaven Roads	" 13	" 13	2	00	19.1
Sou. Drill Grounds	" 13	" 17	3	45	40.7
Hampton Roads	" 17	" 24	3	59	50.9
Wilmington, N.C.	" 25	" 28	22	13	362.5
Sou. Drill Grounds	" 29	" 29	20	05	412.8
Boston	" 31	Nov. 10	35	55	523.1
Rockland	Nov. 10	" 13	7	45	155
"	" 13	" 15	8	29	154.9
"	" 15	" 19	6	24	128.8
"	" 19	" 20	7	24	130.3
Boston	" 21	"	16	08	156.3

TABLE No. 11.

CRUISING REPORT - BATTLESHIP

Port Visited	Arrived (Date)	Departed (Date)	Time Underway Hours	Minutes	Distance Steamed (Knots)
Bremerton, Wash.	-----	Nov.17/22			
Port Angeles "	Nov.17	11/18	7	20	90.4
San Pedro	11/22	11/23	96	18	1199.1
"	11/23	11/24	10	25	135.3
"	11/24	11/27	8	28	59.7
"	11/27	11/28	7	46	73.4
"	11/28	11/29	7	40	55.3
"	11/29	Dec. 1	7	40	50.2
"	Dec. 1	12/ 5	7	46	67.2
"	12/5	12/6	7	56	55.8
"	12/6	12/7	10	46	91.4
"	12/7	12/8	7	38	63.2
"	12/8	12/11	7	51	73.1
"	12/11	12/14	6	53	50.1
"	12/14	Jan. 15	13	40	116.1
"	Jan.15	1/16	5	39	46.7
"	1/16	1/17	6	05	56.8
"	1/17	1/18	7	34	54.2
"	1/19	1/23	12	-	86.2
"	1/24	Feb. 5	12	-	85
"	Feb. 5	2/6	9	-	95.9
"	2/6	2/8	5	-	77.1
Port Culebra,C.R.	2/21	2/23	311	54	3086.6
Panama, C.Z.	2/26	Mar.12	66	37	722.7
"	Mar.15	3/15	32	53	372
"	3/16	3/19	31	08	457
"	3/19	3/23	8	33	113.2
"	3/23	3/31	6	7	67.2
San Pedro	April 11	April 19	205	54	2863.9
"	4/19	4/20	9	-	83.3
"	4/20	May 2	2	48	15.9
"	May 2	5/3	5	06	68.9
"	5/3	5/4	8	12	87.7
"	5/4	5/8	12	6	116.5
"	5/8	5/10	13	-	160.2
"	5/10	5/11	9	6	96.9
"	5/11	5/17	11	42	159.6
"	5/17	5/18	4	6	42.2
"	5/18	5/22	4	48	63.7
"	5/22	5/23	10	06	12.3
"	5/23	5/25	7	24	97.5
"	5/25	5/28	8	12	110.5

TABLE No. 11.

CRUISING REPORT - BATTLESHIP
(Continued)

Port Visited	Arrived (Date)	Departed (Date)	Time Underway Hours	Time Underway Minutes	Distance Steamed (Knots)
Santa Barbara	5/28	5/31	8	24	110.3
San Francisco	June 1	June 25	25	06	335.2
Seattle, Wash.	6/28	July 2	77	42	843.9
Tacoma	July 2	7/9	2	30	29.9
Port Angeles	7/9	7/10	8	24	84.5
"	7/10	7/11	8	14	93.0
"	7/11	7/17	7	25	92.8
"	7/17	7/18	8	-	83.1
"	7/18	7/20	7	39	86.6
Seattle, Wash.	7/20	7/30	6	55	92.2
Port Angeles	7/30	7/31	7	20	93.7
"	7/31	Aug. 1	7	35	80.0
"	Aug. 1	8/3	6	58	71.9
Tacoma	8/3	8/13	8	8	114.7
Port Townsend	8/13	8/14	5	34	73.5
"	8/14	8/15	6	3	76.6
"	8/15	8/17	8	10	79.6
Seattle.	8/17	8/23	3	54	48.6
"	8/23	8/27	9	46	136.5
San Francisco	8/31	Sept.20	99	39	1135
Hunters Pt.	Sept.20		1	30	6.6

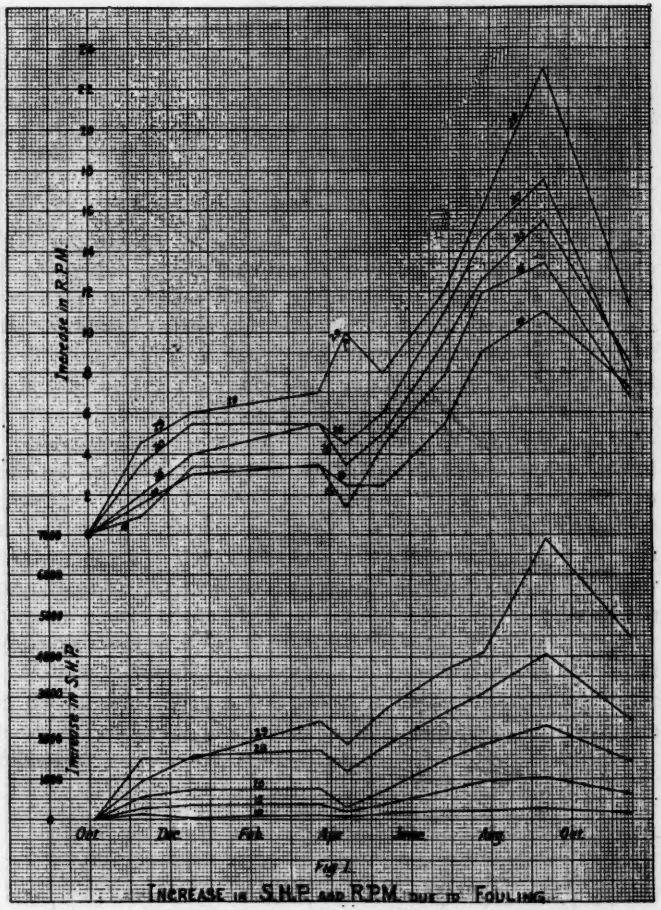


Fig 1.

INCREASE IN S.H.P. AND R.P.M. DUE TO FOULING.

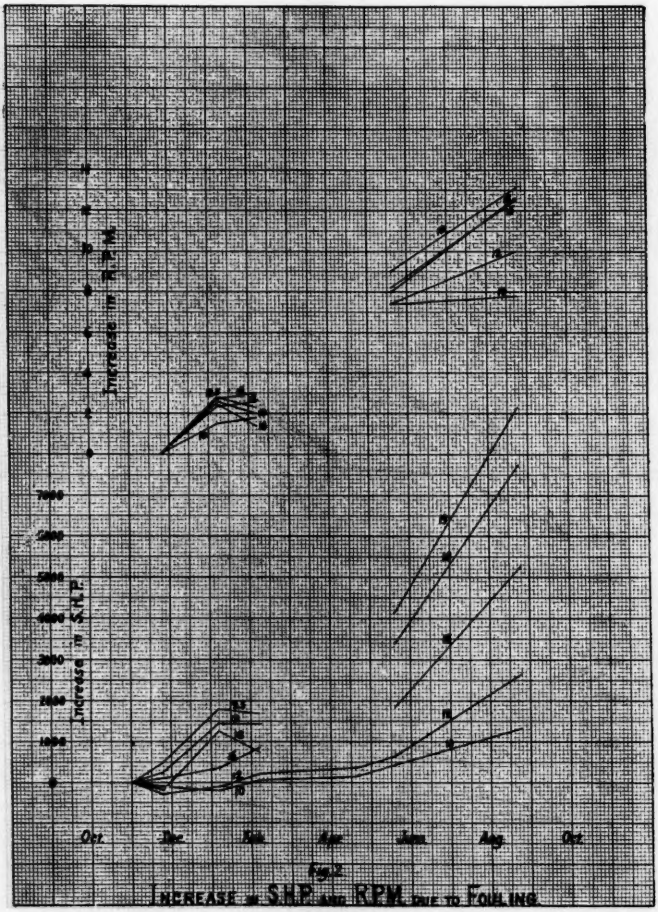


Fig. 12.

INCREASE IN S.H.P. AND R.P.M. DUE TO FOULING.

APPROXIMATE HEAT AND WATER LOSSES.

PAGES 621-28 OF THE NOVEMBER NUMBER OF THE A. S. N. E. JOURNAL.

COMMENT BY COMMANDER H. F. D. DAVIS, U. S. NAVY, MEMBER.

It goes without saying that every possible effort should be made to reduce the heat losses, due to radiation and conduction, from steam piping and machinery to the surrounding metal and air of the machinery spaces. It should be noted, however, that the method used by the author in presenting heat losses in the engine rooms of the Light Cruisers overemphasizes both the amount and relative importance of these losses.

Assuming that the losses of heat in one engine room, as summarized on p. 623, are correct, it follows that the total loss at full power is 74,890 B. T. U. per minute. This is equivalent to approximately 4 pounds of fuel oil per minute. At 80 per cent boiler efficiency, this represents 5 pounds per minute or 300 pounds of oil per hour at the burners of the boilers. The comparative figures at the other powers noted are in proportion to the losses as given in B. T. U. in the table.

Making the further assumption that each horsepower developed in the turbines requires one pound of oil per hour at the boiler burners, instead of the comparison on p. 629, may be written the following:

		20 K.	28½ K.	32 K.	F. P.
(1)	Power developed, H. P.....	6000	20,300	34,400	48,000
(2)	Fuel burned, boilers, lbs/hr.	6000	20,300	34,400	48,000
(3)	Lost in E. R. air lbs/hr.....	241	266	287	300
(4)	Lost in E. R. air, %; (3) ÷ (2) × 100.....	4.0	1.3	0.83	0.7

Continuing the comparison on this basis somewhat farther:

(5)	(1) in lbs. fuel per hour.....	804	2720	4610	6430
(6)	(5) ÷ (2) × 100.....	13.4	13.4	13.4	13.4
(7)	Total loss (2) — (5).....	5196	17,580	29,790	41,570
(8)	(7) ÷ (2) × 100.....	86.6	86.6	86.6	86.6
(9)	(3) ÷ (7) × 100.....	4.64	1.51	0.96	0.72

The assumption that the pounds fuel per H. P. per hour is equal under all conditions makes the "overall efficiencies," line (6), and the percentage losses, line (8), equal throughout instead of showing an economical speed. If, however, the trial consumptions were used, the values of line (9) would be changed but little.

Line (9) shows that the heat losses in the engine room are only a small percentage of the total heat loss. Also this comparison emphasizes the well known fact that most of the heat "goes up the stack" literally and figuratively. In order to determine where effort can best be expended in reducing heat losses and in improving efficiency, the whole field of losses, lines (7) and (8), should be carefully surveyed. Then the most promising area should first be selected for cultivation.

COMMENT BY LIEUT. COMDR. J. K. ESLER, U. S. NAVY, MEMBER.

It is purely a matter of opinion as to the emphasis which should be placed on various kinds of heat losses. In the remarks under consideration the use of 4 per cent of fuel burned at 20 knots to heat the air in one engine room appears sufficient to warrant attention.

There is no question but that when boilers are operating at 80 per cent efficiency the loss "up the stack" or 20 per cent of the fuel burned, is surely the greatest single item of heat loss and warrants the most painstaking effort towards its reduction.

Consideration must be given to relative ease of preventability, and if on an analysis of heat losses these be arranged in order, the most practically preventable at the top and the least at the bottom, I believe it will be found that heat losses by radiation head the list

and heat losses "up the stack" are near to if not at the foot of the list.

The original article on this subject was prepared not so much with the intention of showing actual amount of heat loss by radiation as with the intention of showing that it was appreciable, that it was to a great extent preventable, and to illustrate a method whereby comparisons could easily be made to indicate relative amount of heat losses from radiation in vessels of the same class, or in one vessel at different times.

NOTES.

INDEX TO NOTES.

NAVAL CONSTRUCTION IN 1929
The Engineer, Jan. 3, 1930.

PROGRESS IN MARINE ENGINEERING.
Shipbuilding and Shipping Record, Jan. 1, 1930.

MARINE OIL ENGINE PROGRESS IN 1929.
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THE PROPULSION OF FAST TRANSATLANTIC LINERS.
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ACCUMULATORS FOR USE ON SHIPBOARD.
The Shipbuilder, Dec., 1929.

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Engineering, London, Oct. 25, 1929.

RECORDING FAST OSCILLATIONS.
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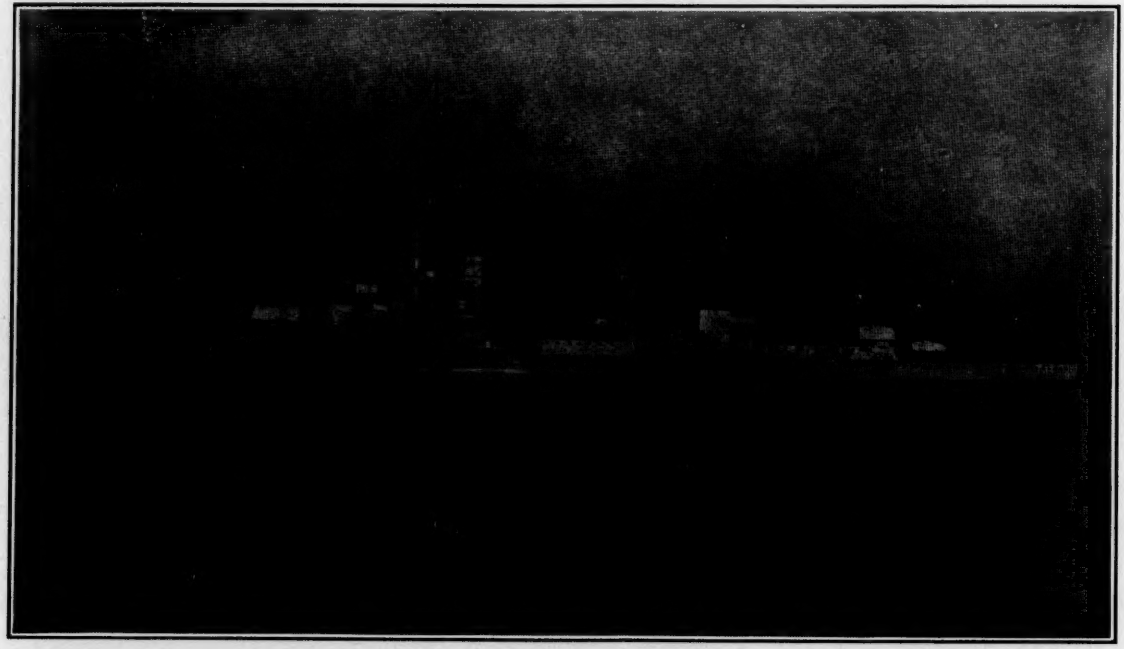
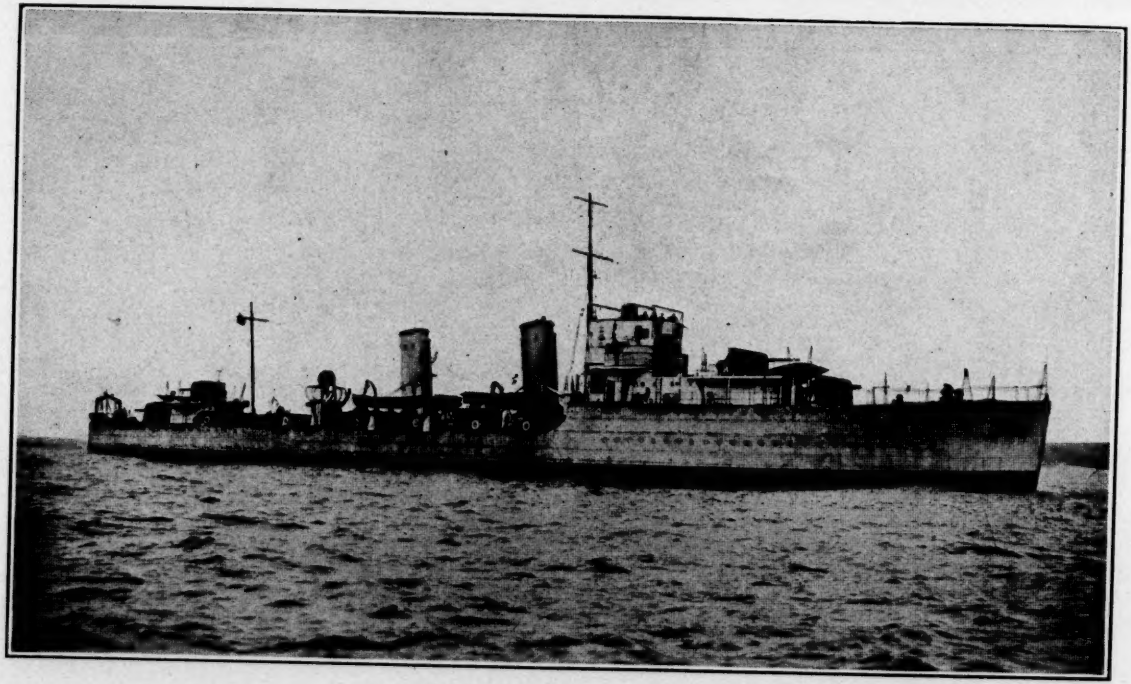
U. S. S. "Salt Lake City."

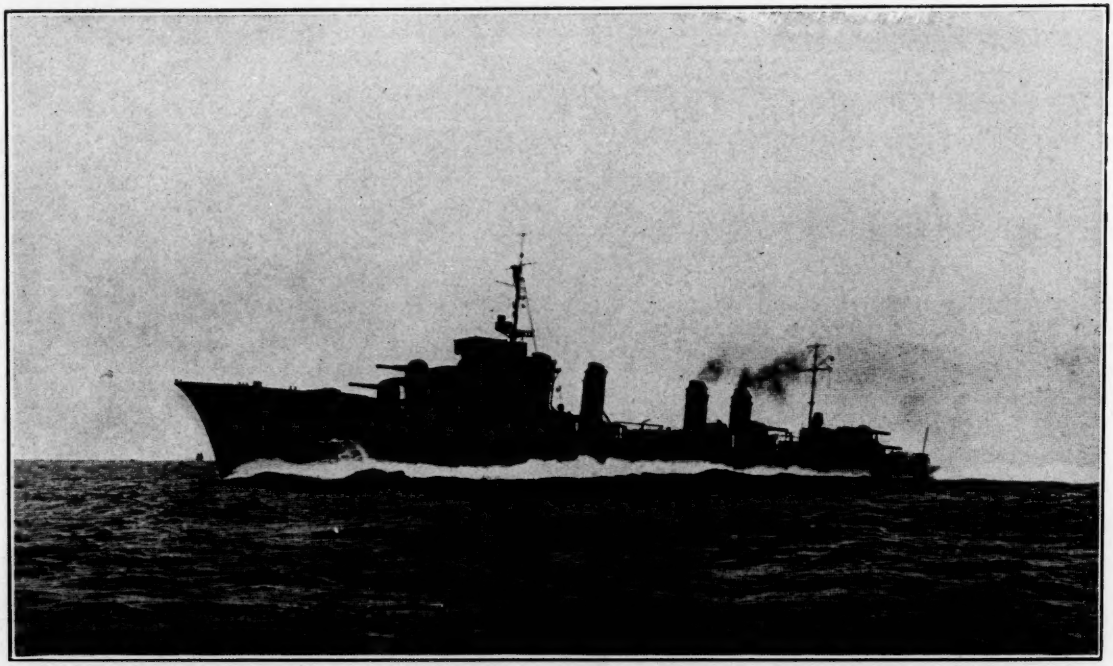


GERMAN LIGHT CRUISER "KARLSRUHE."
(Marine Rundschau).

(Marine Rundschau).



British Destroyer "Active."
(Shipbuilding and Shipping Record).



FRENCH FLOTILLA LEADER "VERDUN."
(Shipbuilding and Shipping Record).

NAVAL CONSTRUCTION IN 1929.

GREAT BRITAIN.

Only a meager amount of new construction was undertaken for the British Navy in the past year. No ship heavier than a flotilla leader was begun, and launchings were confined to two cruisers, a flotilla leader, seven destroyers, and six submarines. Two 10,000-ton cruisers of the 1928 program, *Northumberland* and *Surrey*, were to have been laid down at Devonport and Portsmouth, respectively, but in July it was announced that the construction of these vessels would be deferred, pending the outcome of the Naval Disarmament Negotiations. At the same time a submarine depôt ship, *Maidstone*, which was to have been built at Chatham, and two submarines for which contracts had been placed with Beardmores and Cammell Laird, were cancelled outright. It must be confessed that the outlook for Naval shipbuilding in this country is by no means encouraging. Unless the Five-Power Conference which is to meet in London on January 21st proves a complete failure—which is, to say the least, unlikely—the chances are that no new capital ships or heavy cruisers will be in demand for years to come. The postponement of battleship replacement until 1936, instead of to the year 1931 specified in the Washington Treaty, is generally accepted as assured. It is significant that neither Great Britain nor Japan, nor even the opulent United States, shows any eagerness to resume the building of 35,000-ton ships at a cost of about £8,000,000 per ship. As for the 10,000-ton cruiser, this type has declined in favor everywhere save in America. In spite of its imposing size, speed, and armament. it is dangerously wanting in protection, and may be said to repeat, in an aggravated form, the worst defects of our pre-war battle-cruisers. It has the further drawback of being inordinately expensive. Treaties or no treaties, the British Empire will always, within the range of human calculation, need an abundance of cruising ships, but we cannot afford to build numerous ships at a cost of £2,000,000 per keel. We should not be surprised if the *Dorsetshire*, which was launched on January 29th, 1929, proved to be the last 10,000-ton cruiser built for the British Navy. Including the two Australian units, we now have thirteen of these vessels afloat. They are handsome and, within limits, efficient ships, the design of which does credit to the Admiralty constructors. They can do everything but stand up to heavy punishment, but it is this unavoidable lack of adequate armoring, coupled with their excessive cost, that renders them, in our judgment, unsuitable for British requirements. The Admiralty is known to have in view a new type of cruiser, displacing by "standard" measurement about 6000 tons, mounting 6-inch guns, and costing not more than £1,200,000. If the Government's tentative proposals for cruiser limitation are endorsed at the forthcoming Conference, all, or nearly all, our future cruisers will be of this type. We would observe, in passing, that France, Italy, and Japan show signs of reverting to a smaller type of cruiser, and that even in the United States the expediency of building ships of 6000 to 7000 tons is being favorably considered.

Of the *Dorsetshire*, the only British 10,000-ton cruiser to be launched in the past year, little need be said. Except in minor details, she is a copy of the *London* class, which has been fully dealt with in our columns on more than one occasion. Her sister ship, *Norfolk*, went afloat on December 12th, 1928. Their principal characteristics are:—Length overall, 633 feet; breadth, 66 feet; draught, 17 feet; standard displacement, 10,000 tons; full load, 13,630 tons; machinery, geared turbines developing 80,000 S.H.P., eight small-tube, oil-burning boilers; the designed speed is 32.25 knots. With all bunkers

(Shipbuilding and Shipping Record).

filled, the steaming radius at economical speed is 10,400 miles. Armament—eight 8-inch guns, four 4-inch A.A., eight torpedo tubes on quadruple carriages. Protection appears to be limited to a 3-inch deck over vitals. Owing to the small margin of weight available, the gun turrets are very lightly built, and could not withstand direct hits. Particulars of sub-surface protection are lacking, but in the *Dorsetshire*, as in the *London* group, the external bulge worked into the five *Kents* has been dispensed with.

The *Exeter*, launched July 18th, 1929, and the *York*, which took the water twelve months earlier, represent a modified type which, but for the new disarmament proposals, would probably have been multiplied. The length is 575 feet overall; breadth, 57 feet; draught, 17 feet; displacement, 8400 tons. Although considerably lighter than the *County* class, they have machinery of equal power, and the designed speed of 32.25 knots will, no doubt, be comfortably exceeded. They mount six 8-inch guns, two of the twin turrets being forward and the third aft. In contrast to the *Counties*, there are only two funnels, as the uptakes from the foremost group of boilers are trunked with those of the second group into one large funnel set well abaft the bridge. Both *York* and *Exeter* are unofficially reported to carry two seaplanes mounted on catapults, one on "B" turret, the other on deck abaft the second funnel. If this report is accurate, they will be the first British cruisers to be equipped with a pair of flying machines. For the present, no other cruisers are under construction or projected, but to provide vessels capable of performing the police duties for which the Royal Navy is responsible a number of sloops have been built or are in hand. The first pair, *Bridgewater* and *Sandwich*, launched by Hawthorn, Leslie and Co., in 1928, are now in commission. They are 266 feet long, 34 feet broad, draw 8½ feet, and displace 945 tons. The turbines are of 2000 S.H.P. for a speed of 17 knots. Two 4-inch guns are mounted. The living quarters and the ventilation are specially designed with a view to the comfort and health of the crew in tropical waters. The objection to these vessels is their very small tonnage. Six further sloops were begun in 1929, three at Devonport, one at Chatham, and two by Swan, Hunter and Wigham Richardson, Ltd. In these the displacement has been increased to about 1200 tons. Four more vessels of the same class are to be built under the current Navy Estimates.

The flotilla leader *Codrington*, the first of her type to be laid down since the war, was launched by Swan, Hunter and Wigham Richardson on August 7th. Length, 332 feet; breadth, 33 feet 9 inches; displacement, 1520 tons; geared turbines transmitting 39,000 S.H.P. through two shafts; Yarrow boilers with a working pressure of 300 pounds; contract speed, 35 knots. Armament, five 4.7-inch guns, eight torpedo tubes on quadruple mountings. Allowing for the difference between "standard" and "normal" displacement, the *Codrington* is approximately of the same tonnage as our earlier leaders of the *Shakespeare* and *Scott* class. It will be seen that the Admiralty has elected not to follow the example of France, where leaders of 2700 to 3000 tons are being built in large numbers. The *Keith*, a sister vessel to *Codrington*, has been laid down by Vickers-Armstrongs at Barrow. The seven destroyers launched during the year—*Acasta, Achates, Active, Antelope, Anthony, Ardent*, and *Arrow*—all at Northern yards, are practically uniform with the Thornycroft vessel *Amazon*, completed in 1926. They have the same dspilacement, viz., 1330 tons, but the dimensions have been slightly modified and the contract speed reduced from 37 to 35 knots. The armament of four 4.7-inch guns remains the same, but eight tubes are mounted instead of six as in the *Amazon*. The eighth destroyer of this group, *Acheron*, building by Thornycrofts, will go afloat in a few weeks. Eight further destroyers, comprising the *B* class, with *Beagle* as nameship, were laid down in pairs

during 1929 by John Brown and Co., Hawthorn, Leslie, Palmers, and Swan, Hunter. Of the seven submarines launched in the past year, all save *Orpheus* —February 26th—belong to the *P* or *Parthian* class. They are improved versions of the *O* design, the surface tonnage having been increased from 1540 to 1570 tons, the internal combustion engines from 2700 to 4400 horsepower, and the surface speed from 15 to 17.5 knots. On the other hand, the armament of one 4-inch gun and eight torpedo tubes, of which six are in the bows, is unchanged. Both the *O* and the *P* boats, but particularly the latter, appear to be most successful designs, admirably adapted to the needs of overseas defense.

One of the most interesting naval craft to be completed during the year is H.M.S. *Medway*, a submarine depot and repair ship, laid down by Vickers-Armstrongs at Barrow in April, 1927, and launched in July of the following year. She is shortly to proceed to the China station in company with a flotilla of *O* class submarines. She is virtually a floating base for submarines, and many novel features suggested by war experience find a place in her design. Her dimensions are:—Length overall, 580 feet; breadth, 85 feet; draught, 23 feet; displacement, 15,000 tons. She is propelled by twin-screw double-acting M.A.N. Diesel engines with an output of 8000 B.H.P., the contract speed being 16 knots. The *Medway* is the first auxiliary ship to be provided with special underwater protection, and the first to carry an armament of six 4-inch guns, four of which are on A.A. mountings. She has extensive mooring facilities for submarines lying alongside, including propeller guards and massive floating fenders. In addition to her own fuel oil supply of 530 tons, she can stow in her double bottom 1900 tons of oil for attached submarines. The internal layout of the vessel includes twelve workshops, cold storage chambers of 10,000 cubic feet capacity, and magazines for torpedoes, war-heads and ammunition. The total berthing space is sufficient for 135 officers and 1600 men, every officer being provided with a single-berth cabin. The *Medway* may, therefore, be described as the last word in submarine mother ships. In view of the arduous and trying nature of duty in submarines, more vessels of this type are urgently needed, especially on foreign stations, and it is much to be regretted that the construction of the second ship, *Maidstone*, should have been cancelled by the Government.

Another important fleet auxiliary, the repair ship *Resource*, launched by Vickers-Armstrongs at Barrow on November 27th, 1928, was delivered in the past year. A turbine steamer of 13,500 tons, with a speed of 15 knots, she is fully equipped for all except the heavier categories of repair work afloat. Details of this ship are still scanty, but we understand that all the auxiliary machinery on board is operated by oil-electric power.

UNITED STATES.

Five of the eight cruisers authorized by the Act of December 18th, 1924, were put afloat in 1929, beginning with the *Salt Lake City*, on January 23rd, and the *Pensacola*, on April 25th. These two, which differ materially from the other six, have already been described in "The Engineer," but it will be convenient to repeat their main characteristics. Length overall, 585 feet 6 inches; breadth, 64 feet; mean draught, 17 feet 5 inches; displacement, 10,000 tons; geared turbines of 107,000 S.H.P., four screws, 32.5 knots, White-Forster boilers. Both ships were launched with their main armament on board. This consists of ten 8-inch 55-caliber guns in four turrets, disposed as follows:—A two-gun turret on the forecastle deck, with a three-gun turret in rear of it and superposed, and a corresponding arrangement aft. In addition, four 5-inch A.A. guns and six torpedo tubes are mounted. Protection is

exiguous, for it does not appear that any armor thicker than 1½ inches has been worked into the ship, either vertically or horizontally. American Naval experts acknowledge these ships to be overweighted with armament, and in the six later vessels only nine 8-inch guns are mounted, in three triple turrets, two forward and one aft. Five more cruisers of the same general type were ordered in 1929, and although three of these vessels have been suspended, in consequence of the disarmament negotiations, it is positively reported that the guns and gun mountings for all five ships are in process of manufacture. The only other naval craft to be launched in 1929 were the submarines *V 5* and *V 6*, each of 2760 tons surface displacement, with a speed of 17 knots, and an armament of two 6-inch guns and six torpedo tubes. Excepting the French *Surcouf*, they are the largest submarines built for any navy to date. The program of battleship reconstruction is being carried out according to plan. Up to the present the *Utah, Florida, Arkansas, Wyoming, New York, Texas, Oklahoma,* and *Nevada* have been modernized, and there are now, or shortly will be, in hand the *Pennsylvania, Arizona, New Mexico, Idaho* and *Mississippi*. The work includes bulging, the fitting of extra armor protection, conversion of coal-burning boilers to oil firing, and in the five last-named ships, increase of turret gun elevation from 16 degrees to 30 degrees or 35 degrees. The cost varies from £1,200,000 to £2,000,000 per ship.

JAPAN.

No large ship was launched in 1929, but four 10,000-ton cruisers were completed, the first of their type to be commissioned for the Japanese Navy. *Nachi, Myoko, Ashigara,* and *Haguro* are comparable to the American *Salt Lake City* class, in that they carry ten 8-inch guns and can steam at 32.5 to 33 knots. They have the characteristic "wave-lined" hull, trunked funnels, and towering bridgework of the earlier but smaller 8-inch gun cruisers. Protection is said to be very good, a 4-inch belt in conjunction with steel decks and a triple underwater hull, though it is difficult to believe that all these features have been incorporated in a 10,000-ton ship with such heavy armament and high-powered machinery. Four further vessels of the same type are under construction, bearing the names of *Atago, Takao, Chokai,* and *Maya*. Upon the completion of these ships Japan will possess four 7100-ton and eight 10,000-ton cruisers, mounting in all 104 guns of 8-inch caliber. The aircraft carriers *Kaga* and *Akagi*, rebuilt from capital ship hulls, have both been delivered. They bear a superficial resemblance to H.M.S. *Furious*, except in the arrangement of the ducts for discharging furnace smoke and fumes. The *Kaga* has enormous flues, which are carried externally on either side of the ship towards the stern, where they are turned outboard; the *Akagi* has smoke ducts which are simply huge funnels trunked outward and downward amidships. A much smaller aircraft carrier, the *Ryujo*, has been ordered from the Yokohama Dock Company. She will displace only 8100 tons, and is, apparently, to be a reduced copy of the *Hosho*. Several destroyers were launched during the year, all belonging to a group of twenty-four craft, *Fubuki* class, authorized in 1926. They are large vessels, 368 feet in length, displacing 1700 tons, with a contract speed of 35 knots, and are armed with six 4.7-inch guns and nine tubes. Their most noticeable feature is that their guns are paired in gas-tight light splinter-proof gun-houses, and they are believed to be the first destroyers in which guns have been so mounted. A submarine, *I 60*, which was launched at Sasebo on April 24th, is typical of the heavy, fast, ocean-going class which has been recently developed in the Japanese Navy: length, 330 feet; surface displacement, 1650 tons; surface speed, 21 knots; armed with one 4.7-inch and one 3-inch gun

and eight tubes. These vessels are remarkable for their high speed and the radius of action is also reported to be unusually wide. Other ships launched in 1929 were three anti-submarine net layers, one of 1345 tons and 16 knots, the other two of 450 tons and 19 knots. They are the first specially designed vessels of this type to be built in any country.

FRANCE.

Alike in the number and aggregate tonnage of warships launched, France was far ahead of other Powers in 1929. The principal vessels floated were the *Foch*, a 10,000-ton cruiser, at Brest on April 24th; *Commandant Teste*, a seaplane carrier or tender, of 10,000 tons, at Bordeaux on April 12th; and the *Pluton*, a cruiser minelayer of 5300 tons, at Lorient on April 10th. Several destroyers and half a dozen submarines also took the water. The *Foch* belongs to the conventional type of 10,000-ton *Treaty* cruiser, but in contrast to the earliest representatives of this class, *Duquesne* and *Tourville*, the speed has been reduced and protection improved. A sixth ship of this generic type, *Dupleix*, was begun in November, and a seventh has just been authorized. The *Commandant Teste* is in some respects a hybrid, her nearest of kin being the Australian seaplane carrier *Albatross*. In spite of her considerable tonnage the French vessel has no landing deck, the machines she carries being projected by means of catapults. Her speed of 20 knots is inadequate, and the armament of twelve 3.9-inch guns unimpressive. The minelayer *Pluton*, on the other hand, appears to be a very promising design. She is compared below with H.M. cruiser-minelayer *Adventure*, and it must be admitted that the French vessel seems to have the best of the comparison.

	Pluton.	*Adventure.*
Length, overall, feet	472½	520
Breadth, feet	51	59
Draught, feet	17	19¼
Displacement, tons	5300	6740
Speed, knots	30	27.75
Armament	Four 5.5-inch, 250 mines	Four 4.7-inch, 320 mines

The British ship's advantage lies in her greater mine capacity and the installation of an auxiliary internal combustion propelling plant for cruising, but in speed and gun armament she is outclassed by the *Pluton*. The 2780-ton flotilla leaders *Vauban*, *Valmy* and *Verdun* were all completed or launched during the year. By reaching 40.18 knots on her steam trials, the *Verdun* established what is claimed to be a world's record, though it was approached to within a fraction, more than ten years ago, by the British destroyers *Teazer* and *Tyrian*. The new French leaders are handsome and powerful vessels, heavily armed—five 5.5-inch guns, six tubes—with good sea-keeping qualities and ample steaming endurance. They may be regarded as the lineal descendants of the French torpedo-gunboats which caused a flutter on this side of the Channel in the nineties of last century.

The submarine cruiser *Surcouf*, launched at Cherbourg on November 18th, is the largest submersible craft which has yet been built. She is 393.7 feet in length, 29.5 feet in breadth, and draws 23 feet when in surface trim. Her tonnage is 3250 tons on the surface and 4300 tons submerged. The designed surface speed is 17 knots. According to semi-official details, she is to be armed with four 5.5-inch guns in turrets and fourteen torpedo tubes. The deck is armor-plated and the hull so strongly built that the vessel will be able

to dive with safety to a depth of 60 fathoms. A hangar is provided for a small seaplane. The *Surcouf* is credited with a radius of 12,000 miles. She is obviously an experimental type, like the British *X 1* and the American *V 4*, and as such her career will be watched with interest.

ITALY.

No large ship was launched in the past year, but work proceeded on the 10,000-ton cruisers *Zara*, *Fiume*, *Bolzano* and *Gorizia*, which are modified versions of the *Trento* and *Trieste*, completed in 1928-29. The four later ships are slower, 32 knots as against 35.5, and have stronger protection—another example of the universal reaction from the original tendency to design these *Treaty* cruisers as mere speed machines—and 8-inch gun-platforms, without regard to defensive properties. Official details are now available of the six *Condottieri* cruisers, of 5250 tons, which represent a refreshing departure from the more or less stereotyped 10,000-tonners. Their characteristics are: Length, 597 feet; breadth, 51 feet; draught, 14 feet; geared turbines of 95,000 S.H.P. for 37 knots; protection, a 2-inch to 3-inch steel deck and splinter-proof turrets. Armament: Eight 6-inch guns, paired in turrets; four 4-inch A.A. guns, and four torpedo tubes. Once again Italian designers have exhibited their genius for putting extraordinary qualities of fighting power and mobility into vessels of modest displacement. On paper, at all events, these are the most formidable cruisers of their size which have been produced since the war. Among the destroyers launched in 1929 were several of the *Navigatori* class, comprising 12 units. They displace 1908 tons, have a speed of 38 knots, and carry six 4.7-inch guns. Eight smaller destroyers of the *Dardo* class, 1350 tons and 38 knots, have been laid down.

GERMANY

The only naval unit launched during the year was the *Leipzig*, at Wilhelmshaven on October 18th. This vessel is the fifth and last cruiser of the replacement program, and no further cruising ships are projected for the time being. The *Leipzig* and her predecessors have been dealt with in this journal, and it is only necessary to add that the most has been made of the 6000 tons displacement to which their designers were restricted. Excepting the *Emden*, which has eight 5.9-inch, they all carry nine 5.9-inch guns in triple turrets. The speed is 32 knots—*Leipzig* 32.5—and in the *Königsberg*, *Karlsruhe*, *Köln* and *Leipzig* an auxiliary internal combustion plant is installed to drive the ship at cruising speeds. Electric welding has been extensively employed in the building of the hull, and by this means, together with the use of special alloys, a substantial margin of weight has been saved. They are admirable ships, which would do credit to any fleet. Very little new information has come to hand about the 10,000-ton armored ship *Ersatz Preussen*, which was laid down at the Deutsche Werke, Kiel, in September, 1928, and is to be launched in the coming summer. Her leading particulars and official plans were published in "The Engineer" of January 11th, 1929. It is known that her displacement of 10,000 tons is a "standard" figure, and that her load displacement will be nearly 14,000 tons. She is to be propelled by internal combustion engines of 50,000 B.H.P., which are being built in the M.A.N. shops at Augsburg. They will operate twin screws through reduction gearing supplied by the Vulkan yard, Hamburg, the revolutions between motor and propeller being reduced in the ratio of 9 to 5. The net weight of the motors is hardly more than 17.64 pounds per brake horsepower. Al-

though the total weight of the complete plant, comprising motors, gearing, shafting, propellers, accessories, etc., will naturally work out at a higher figure per unit of power, it will still be less than that of a modern geared turbine installation of equal capacity. The contract speed is 26 knots, and at a speed of 20 knots she will, it is asserted, have the surprising endurance of 10,000 miles. The armament is to be six 11-inch guns in triple turrets, eight 6-inch behind shields, four 3.4-inch A.A. guns, and six torpedo tubes. For protection she will have an almost complete belt on the water-line, two decks, and extensive subdivision. When, almost exactly twelve months ago, details of the *Ersatz Preussen* were first made known by "The Engineer," we ventured to remark that the appearance of such a vessel, with its powerful armament and phenomenal radius of action, introduced a new factor into the naval situation and might eventually necessitate a revision of the Washington Treaty rules governing warship design. These observations were discounted in certain quarters, but they have since found widespread acceptance. There is hardly any doubt that the *Ersatz Preussen's* design will materially influence the decisions reached at the London Naval Conference.

OTHER NAVIES.

After lying derelict ever since her launch from the Vulkan yard at Hamburg fifteen years ago, the Greek battleship *Salamis* is at last to be completed and handed over to the Hellenic Navy. Her original design, it is understood, is to be revised, but to what extent is not yet known. She displaces 19,500 tons (normal), and was to have had turbines of 40,000 S.H.P. for a speed of 23 knots. The armament provided for was eight 14-inch guns, twelve 6-inch and twelve 12-pounders, with three submerged tubes. The decision of Greece to take belated delivery of this ship is traceable to the expansion of the Turkish fleet. The latter has recently been strengthened by the *Yavuz Sultan Selim* (ex-German battle cruiser *Goeben*), which was thoroughly refitted at Constantinople by the French firm, Chantiers de St. Nazaire. Moreover, the Turkish Government has under construction in Italian yards two 1350-ton destroyers of 36 knots, and two submarines, of 950 and 880 tons, respectively.

The past year witnessed the completion by British firms of important naval contracts for South America. Of these the largest was that booked by Thornycrofts, late in 1927, for six Chilean destroyers. They were of uniform design, 1090 tons and 35 knots, and all exceeded the contract speed on trial. For the same Government Vickers-Armstrongs have just delivered three submarines of 1540 tons, which closely resemble the British *O* class, and launched in November the submarine depot ship *Araucano*, of 4000 tons gross. Further, Vickers-Armstrongs are building two Chilian naval oil-tankers, each of 3800 tons gross. For the Argentine Government, J. Samuel White and Co., of Cowes, have completed three flotilla leaders of 1520 tons. The designed speed is 36 knots, but the *Mendoza* and *Tucuman*, on trials, maintained 38 knots for 6 hours, and *La Rioja* reached a maximum of 39.4 knots. We understand that the South American Governments concerned are more than satisfied with the warships they have recently had built in this country, and there are some grounds for anticipating the receipt of further contracts from the same source. The Argentine cruiser *Veintecinco de Mayo*, built by Orlando Bros., of Leghorn, was launched on August 11th. She is a vessel of 6800 tons, a designed speed of 32 knots, and a main armament of six 7.5-inch guns. A sister ship is shortly to be launched at the Odero yard, Sestri Ponente.—"The Engineer," Jan. 3, 1930.

PROGRESS IN MARINE ENGINEERING.

PULVERIZED FUEL TO THE FORE—EXHAUST STEAM TURBINE DEVELOPMENTS—PROGRESS WITH ELECTRIC DRIVE—INDEPENDENTLY-FIRED SUPERHEATERS—THE "BREMEN"—NEW LARGE DIESEL ENGINES.

If one were asked what had been the outstanding marine engineering development of 1929, the answer would probably be "pulverized fuel." Since our last annual review of marine engineering progress was written, much has been done to further the successful application of pulverized fuel afloat. During last August, for instance, no fewer than three Scotch-boilered vessels, burning pulverized fuel under their boilers, were inspected by us, each being fitted with a different system of burning the powdered coal. The three ships referred to, it may be recalled, are the *West Alsek*, *Swiftpool* and *Berwindlea*. The first had the distinction of being the first vessel wholly dependent on pulverized coal as fuel to cross the Atlantic without recourse either to supplementary or emergency oil burning. This performance was undoubtedly a definite milestone along the road of progress in the art of burning powdered fuel in a marine boiler. The *Berwindlea* and *Swiftpool* were interesting as being the first new British ships to adopt pulverized-coal firing. Readers will recall that the *Swiftpool* is fitted with the Brand system of pulverized-coal burning, and was fully described and illustrated in our issue of August 1, to which reference should be made for fuller details of the equipment. The *Hororata* is another conversion to pulverized coal that was made last year and is giving a good account of herself. She uses the Howden-Buell system.

The *Berwindlea* is fitted with the Clarke Chapman system, which has the advantage of a fair amount of valuable sea-going experience behind it. This system, in addition to having a fairly extensive land application, had already been fitted in the steamers *Stuartstar* and *Musician*, so that its sponsors were in a particularly good position when they undertook the *Berwindlea's* installation. Consequently, the results achieved were very satisfactory, the extraordinarily high boiler efficiency of 87 per cent being realized on trial and afterwards maintained in service.

As the *Berwindlea's* installation has not been described in our pages, a few details of the equipment might not be out of place. The *Berwindlea* is provided with two single three-furnace Scotch boilers 16 feet 6 inches in diameter by 12 feet 6 inches long, these supplying steam at a pressure of 200 pounds per square inch to a triple-expansion engine capable of developing a service power of about 2000 I.H.P. at 68 R.P.M. Each boiler has a separate motor-driven pulverizer of the Clarke Chapman Resolutor type. This is a simple machine of the beater arm type, which normally rotates at about 1450 R.P.M. The primary hot air fan is driven off the same spindle as is the pulverizer, the fan suction being from the fifth stage of the Howden corrugated plate air heater installed in the boiler uptake. The system of distribution to the three burners is interesting for, instead of any arrangement of mechanical distributor, a ring main device is adopted. As can be seen from the plan of the boiler-room of the vessel, the main supply pipe is of gradually-reducing diameter where it passes the boiler front, the main being so proportioned that the delivery of powdered coal and hot primary air mixture to each burner is approximately the same. Any surplus beyond the requirements of the three burners for the particular conditions is circulated around the ring main back to the suction side of the primary-air fan.

The burners are of the Woodeson short-flame pattern, similar to those used in the *Stuartstar* and *Musician*, the secondary air being obtained direct from the Howden air box.

The *West Alsek* is deserving of brief description, because the Todd system of pulverized fuel firing adopted differs radically from either the Brand or the Clarke Chapman systems used in the *Swiftpool* and *Berwindlea*, respectively. The Todd engineers believe in solving distribution difficulties by avoiding them. Thus, a separate pulverizer is provided for each burner. The vessel has three three-furnace boilers, each boiler having three high-speed pulverizers driven in series by a small impulse type steam turbine.

The general arrangement of the Todd equipment as applied to one boiler is shown in sketch.* The coal is fed by gravity from the bunker into a steam engine-driven crusher. From there it is passed into a screw conveyor which is driven by a variable speed electric motor. This screw conveyor may be driven from either end with a view to trimming the bunkers from port or starboard, as may be desired. The Todd burner presents several interesting features, and is, therefore, also illustrated. It is of the turbulent short-flame type and is designed to give rotation to the coal-air mixture with a view to assisting mixing and combustion. A feature of the design is the high temperature of the primary air. The burner diaphragm plate is backed by a hollow ribbed cast-iron cover plate which is supplied with air from the ordinary Howden front. This air effectively cools the diaphragm plate and hence reaches a high temperature. The heated air is led away from the hollow diaphragm under the boiler-room floorplates to the primary-air fan. Secondary air is introduced to the burner tangentially and meets the strongly rotating coal-air stream, with result that an intimate mixture is ensured and combustion initiated very rapidly.

Yarrow & Co. Ltd. are another firm who have given the pulverized fuel problem, so far as it affects marine work, their very close attention. For some considerable time numerous experiments have been carried out on both Yarrow water-tube and Scotch-type boilers. Various burner designs have been tried out at Scotstoun, and the firm have developed a brick-lined pre-furnace for use with their system. The pre-furnace is mounted on a wheeled frame, so that it may readily be withdrawn from the boiler front for examination and, if necessary, renewal of the brickwork. The Yarrow engineers have found that this pre-furnace is an essential feature to success with their design, for not only does it protect the front riveted seam of the boiler from the intense heat of combustion, but it also improves combustion and renders ash removal easier. It has been found that most of the ash is deposited in the pre-furnace and as special doors are provided at the bottom of the pre-furnace, the ash may be readily removed from time to time. The final form of the Yarrow short-flame burner presents several clever features and is, therefore, deserving of discussion. It consists essentially of an outer tube and an inner tube, pulverized fuel and primary air entering along the pipe shown in the sketch and flowing into the annular passage between the two tubes. This passage has a venturi constriction at an intermediate point, where the inner burner tube is divided transversely, and where also a special nozzle formation is made. The exit of the passage between the burner tubes is fitted with vanes which impart a rotary motion to the powdered coal and primary-air mixture, an adjustable suction tube being fitted inside the end of the inner burner tube. The secondary air passes through the surrounding

* The Todd equipment was described in detail in the May, 1929, number of the JOURNAL A.S.N.E.

heated air box, through direction vanes which give it a rapid whirling motion, in which state it meets the coal-air mixture at the mouth of the burner. Due to the arrangement of the venturi constriction and divided inner tube, the pressure within the inner tube is reduced and results in a backward flow of hot gases from the mouth of the burner, these gases being drawn into the venturi throat to return along the final half of the annular burner passage, thus pre-heating the fuel stream. By means of the adjustable suction tube the amount by which the inner part of the expanding vortex of burning fuel leaving the burner turns back into the mouth of the inner burner tube is controlled. Furthermore, this feature of design also enables the length of the flame to be regulated by diverting the secondary air more into the furnace.

It is interesting to note that the Yarrow system is being applied to the Henderson liner *Amarapoora*. The pulverizing mills to be used on this vessel are of the Atritor type manufactured by Alfred Herbert Limited, Coventry, this type of mill being the one with which Yarrow & Co. Ltd. have had much experience during their experimental work.

EXHAUST-STEAM TURBINE DEVELOPMENT.

During the year under review the exhaust-steam turbine has made great headway, not only for conversion purposes but as a means of improving the economic efficiency of new steamships. The pioneer Bauer-Wach system, which, it will be recalled, is handled in this country by Wm. Beardmore & Co. Ltd., and Swan, Hunter and Wigham Richardson Limited, has continued to make great headway, many notable British and Continental vessels having been provided with this equipment. The largest vessel yet fitted with the Bauer-Wach turbine is the Hamburg-Amerika liner *Cleveland*. She is an oil-burning vessel of about 17,000 tons, and is propelled by two large quadruple-expansion engines. Her total power is now about 13,000 I.H.P. and her service speed around 15½ knots. A most interesting application of the Bauer-Wach system is its adoption for a Rhine paddle tug. The usual quill drive could not be adopted in this instance, the desired degree of flexibility being obtained by means of a Bibby coupling. This is an important application which will be watched very closely by tug owners throughout the world.

In our last review it was mentioned that the Parsons Marine Steam Turbine Company and Brown, Boveri Limited, of Baden, Switzerland, were developing an exhaust-steam turbine drive for use with reciprocating-engined steamers. Up to the time of writing, however, neither of these systems has been applied to a vessel. The Parsons design was dealt with in our last review, but at that time full particulars of the Brown, Boveri type were not available. The latter system is simple and straightforward and possesses various interesting features. The Swiss arrangement differs radically from the Bauer-Wach system, in that the turbine is not uncoupled when the main engine is reversed. In order to facilitate rapid reversing of the turbine, despite its high kinetic energy, one or two rows of astern blades are provided to which live steam is admitted simultaneously with the throwing over of the control lever to astern. When changing back to the ahead position, live steam is likewise admitted to the ahead turbine. The valves for controlling this steam and for operating the by-passing valve by which the steam is diverted either to the condenser or to the turbine are positively actuated by a pressure oil system.

A notable addition to exhaust-steam turbine equipment is the system which the Metropolitan-Vickers Electrical Company have developed in association

with Mr. H. Hinchcliffe, superintendent engineer of the Ellerman Lines. This system was fitted for the first time in the Ellerman liner *City of Hongkong*, a full description of which appeared in our issue of November 14. In view of the fact that this installation was so recently described, it will not be necessary to recapitulate the principal features of this new system which, in passing, is giving excellent results on the *City of Hongkong*. A turbo-electric exhaust turbine system has also been developed by the British Thomson Houston Co. Ltd., Rugby, and has been fitted on a limited number of British-owned steamers. At the time of writing, no data are available regarding either the performance of these vessels or the general arrangement of the equipment. The use of electricity in a supplementary propulsion system of the type under discussion is an interesting development which has not a little to recommend it, for the adoption of the electric drive permits of the necessary speed reduction between generator and propulsion motor being conveniently arranged and, at the same time, the adoption of any form of clutch or fluid coupling is avoided.

While on the subject of the electric drive, a review of this character would be incomplete without some reference to the large P. & O. turbo-electric liner *Viceroy of India*, which went into service in March of last year. This ship, engined by the British Thomson-Houston Co. Ltd., Rugby, has the distinction of being the first large turbo-electric liner to be constructed in this country. The *Viceroy of India*, it will be recalled, is a twin-screw vessel of about 17,000 S.H.P. at 109 R.P.M. of the propellers. Two three-phase 9000 Kw. at 2690 R.P.M. turbo-alternators are provided, the supply pressure being 2720 volts. The main propulsion motors are slow-speed synchronous machines, excitation current for alternators and propulsion motors being obtained from separate machines. Variation of propeller speed in either direction is obtained by varying the turbine speed, while the direction of rotation of the propellers is changed by reversing the connections of two of the three phases leading from the alternators to the motors. This latter operation is accomplished by means of contactors operated by levers located on the control platform. When starting or reversing, the turbo-alternators operate at approximately one-fifth full speed or below, and the propulsion motors are brought up to a corresponding speed as induction motors. The field windings of the alternators are temporarily over-excited during the reversing and accelerating period so as to bring the propulsion motors up to speed and to bring them almost into synchronism. The motor fields are then excited so as to bring the motors fully into synchronism, and finally the alternator field strength is reduced to normal. The *Viceroy of India*, which is provided with Yarrow high-pressure boilers and three-stage feed heating, using bled steam on a Weir closed-feed system, has given eminently satisfactory results since she went into service, the ability to operate at reduced speed on one turbo-alternator and four of her six boilers giving a high degree of economy. Doubtless because of her very satisfactory operation, it has frequently been rumored during the past few months that the P. & O. Company contemplate the construction of further turbo-electric liners. Up to the time of writing, however, no additional large tonnage has been ordered by the Company, although ships for certain P. & O. subsidiaries were recently ordered from Harland & Wolff Limited.

INDEPENDENTLY-FIRED SUPERHEATERS.

One of the most interesting technical developments during the past year in connection with steam tonnage was the fitting of an independently-fired superheater in the Compagnie Générale Transatlantique's new turbine

steamer *President dal Piaz.* This vessel was built and engined by Swan, Hunter & Wigham Richardson Limited, and was completed about the middle of June. Her two sets of single-reduction geared turbines have a total power of about 5000 S.H.P. at 155 R.P.M. of the screws, steam being generated in four oil-fired Prudhon-Capus combined fire-tube and water-tube boilers. From the boilers the steam is passed into the independently-fired superheater, which is located between the after pair of main boilers.

This superheater was designed and built by the Superheater Co. Ltd., London and Manchester. It is fired on the Wallsend-Howden system, as are also the boilers of the vessel. Thermostats are provided in the steam outlet branch of the superheater and also in the superheater furnace, and should either steam temperature or gas temperature rise unduly high, the thermostat operates on the oil-burning installation pumps so as to regulate the supply of fuel to the burners. In this ingenious manner the superheater is protected against over-heating and the turbines safeguarded against damage from unduly high steam temperature. The superheater embodies recuperative gas control, a large proportion of the gas leaving the superheater being returned down the back of the casing of the furnace by a fan used to reduce the temperature of the combustion gases before they reach the superheater. In this way the oil can be consumed with a high percentage of CO_2 instead of introducing a large amount of excess air. Thus, the efficiency of the superheater is high—over 90 per cent—and the fuel consumption moderate. The arrangements for baffling the gases as they pass through the superheater are also interesting. The superheater elements are arranged in such a way that at intervals they lie side-by-side across the superheater chamber and form their own gas baffles. Consequently, the usual brick and iron baffling is dispensed with. By regulating the oil fuel supply and the dampers controlling the return of the gases to the superheater and to the funnel, the temperature of the steam can be maintained within very close limits over a wide range, while the superheater can also be cut out of action at a moment's notice when required. This development is an interesting one which undoubtedly has some very good features to recommend it, and it will be interesting to see whether the bold experiment of the Compagnie Générale Transatlantique will be repeated by other steamship companies.

One of the most notable steam-propelled vessels of the year under review is undoubtedly the Holland-Amerika liner *Statendam,* a twin-screw liner of approximately 30,000 gross tons and 19 knots service speed. She is propelled by two sets of single-reduction geared turbines of the three-casing type, these being capable of developing an aggregate power of 20,000-22,000 S.H.P. at 125 R.P.M. The ship is noteworthy in having high-pressure boilers, superheated steam being supplied at a pressure of about 430 pounds per square inch and 650 degrees F. by six Babcock & Wilcox boilers which are oil-fired on the White system. Electrically-driven pumps are used for a number of services in the engine-room, the necessary current being supplied by four 400-Kw. Diesel generator sets. By carefully-designed multiple-stage feed heating and the minimizing of latent heat of evaporation losses for the installation as a whole, by the use of special feed heating arrangements, using steam for cooking, heating, etc., the overall thermal efficiency of the plant is remarkably high; in fact, the *Statendam* has the distinction of being the most economic steamship in the world, a specific fuel consumption of 0.61 pound per S.H.P. per hour being returned in service at the end of the liner's first trip.

While much of the credit for this fine performance is very naturally Dr. Meyer's (the superintendent engineer of the line), Harland & Wolff Lim-

ited, who built and engined the vessel, are also deserving of warm commendation for their efforts in this connection. For one of the foremost specialists in the design and construction of motorships to achieve such a noteworthy result with a steam vessel is a very fine performance, the significance of which has perhaps not generally been realized. This should do much to explode the erroneous popular idea that the Belfast firm are motorship specialists first and last.

Unquestionably, the "ship of the year" is the Norddeutscher Lloyd express liner *Bremen*, the 49,000-ton quadruple-screw vessel which the Deutsche Schiff- und Maschinenbau A.G., Bremen, built to capture the Blue Riband of the Atlantic. The circumstances in which this fine new vessel attained her designers' and owners' objective are too fresh in everyone's memory to call for more than passing reference in this review. It is sufficient to say that on her maiden voyage the *Bremen* made the passage from Bremerhaven to New York, a distance of 3164 miles, at an average speed of 27.83 knots. The return trip was made at an average speed of 27.91 knots. The propelling machinery consists of four sets of single-reduction geared turbines of the reaction type, each set consisting of high, intermediate and low-pressure ahead units with independent astern turbines. All turbine casings are of cast steel and all rotors run at 1800 R.P.M., this being reduced to a propeller shaft speed of 180 R.P.M. One of the most interesting features of the machinery is the unusual and, in our opinion, very ingenious pinion design. The torque of the turbine is transmitted directly to a flange in the center of the pinion, and from there equally to the two tooth faces. This arrangement minimizes torsional deformation of the pinion and equalizes it between the two halves, thus entailing the adoption of a sleeve shaft or quill drive. Steam is supplied at a working pressure of 330 pounds per square inch by 20 oil-fired water-tube boilers of a modified Thornycroft type. Eleven of these are double-ended and nine single-ended, the boilers being arranged in two widely-separated compartments. The closed stokehold system of forced draught is used. The steam is superheated to a temperature of 700 degrees F. in superheaters incorporated with the boilers. Electrical energy for driving the various deck machinery, engine-room pump motors, etc., is supplied by four Weser-M.A.N. 520-Kw. Diesel generator sets, which are installed in an auxiliary engine room. Although the service speed of the *Bremen* is about 28 knots on 100,000 to 110,000 S.H.P., a trial speed of 28½ to 29 knots was obtained on 120,000 to 130,000 S.H.P. It is believed, however, that this latter figure is rather below the maximum of which the machinery is capable.

DIESEL ENGINE DEVELOPMENT.

Since our last review was written some notable motorships have been put into service, both by British and foreign owners. These include such interesting ships as the Swedish-Amerika liner *Kungsholm;* the New Zealand Shipping Company's liner *Rangitiki;* the motor-tanker *Irania,* the first vessel with the new Richardsons Westgarth double-acting engine; the *Aldington Court,* the first motorship for the well-known Court Line; the large Krupp-engined tanker *Calgarolite;* the cross-Channel motorship *Ulster Monarch;* the geared motorships *Milwaukee* and *St. Louis;* the British-built trunk-piston-engined cargo motorship *Villanger;* the Blue Funnel supercharged motor liner *Agamemnon;* and, recently, the Union-Castle liner *Llangibby Castle.*

Turning to Diesel-engine developments, to our mind the most outstanding development of the year is undoubtedly the appearance of the Burmeister & Wain two-stroke cycle double-acting engine. Details of this interesting new engine were first published in the paper which Mr. H. H. Blache, technical director of Burmeister & Wain, read before the Engineering Congress at the Centenary Celebrations in connection with the Royal Technical University, Copenhagen. An abstract of Mr. Blache's paper was published in our issue of October 3, but in view of the unique features of the new engine, we reproduce in another column a section through the cylinder of the Burmeister & Wain two-stroke engine. Instead of utilizing the usual port scavenging arrangements, a straight-through scavenge is employed, upper and low exhaust ports being controlled by means of piston valves. These piston valves are connected by a system of side rods and drive a small auxiliary crankshaft located inside the crank chamber. This shaft is connected to the main crankshaft by means of a chain drive and it is stated that the small auxiliary pistons develop approximately 10 per cent of the total power of the engine. Thus, the valve gear is not only "self-supporting" but returns a useful amount of work to the main crankshaft and so enhances the mechanical efficiency. As airless injection is employed, the mechanical efficiency will, in any case, be high. A six-cylinder engine of this new type will shortly be in service in the East Asiatic Company's motorship *Amerika*. This engine develops 7000 B.H.P. at 100 R.P.M., the cylinders being 620 millimeters in diameter by 1400 millimeters stroke. The total weight of the engine is said to be about 380 tons, which is a considerable reduction on that of the corresponding Burmeister & Wain four-stroke cycle double-acting engine.

During the year under review, Burmeister & Wain's airless-injection system was adopted on a number of important vessels, including the *C. F. Tietgen;* the Blue Funnel liner *Agamemnon;* and the Belfast-Liverpool service motorships of the "Ulster Monarch" class. Engines of this type have also been constructed by the Stabilimento Tecnico Triestino. Burmeister & Wain's airless-injection system is simple and straightforward. A separate fuel pump is provided for each cylinder, the pumps being driven from the camshaft and located as near as is practicable to the fuel valve which they serve. The fuel valve is an automatic differential plunger type of simple and conventional formation. It is understood that the system, which is of the spill valve type, has given eminently satisfactory results, and in view of the simplification and reduction of first cost which it makes possible, there is not the slightest doubt that it will become increasingly popular during the coming year.

A PROMISING ENGINE.

Publicity has very rightly been given to the Richardsons-Westgarth two-stroke cycle double-acting airless-injection engine which, at the time of writing, is the only all-British design of this type available to the shipowner. The engine is one which displays considerable ingenuity and originality. Ease and cheapness of manufacture, without impairing efficiency or reliability, has been the aim of the designer; and the performance of the first vessel to be fitted with this engine, the small motor-tanker *Irania*, indicates that Mr. W. S. Burn has gone a long way towards achieving the desired end. Specific fuel consumptions of under 0.40 pound per B.H.P. per hour are being obtained from this first installation which, now that its teething troubles have been overcome, is operating with great reliability. Mr. Burn

read an interesting paper before the North-East Coast Institution of Engineers and Shipbuilders some little time ago in which he gave the results of his later work in the development of this new engine, together with the service performance of the *Irania*. This paper was given in abstract form in our issue of November 14, to which reference should be made for fuller information.

During 1928 the Hamburg-Amerika Line put two interesting motor vessels into service in the *St. Louis* and *Milwaukee*. Both are propelled by four high-speed two-stroke cycle double-acting M.A.N. engines, driving two propeller shafts through single-reduction gearing. In the case of the *St. Louis*, the Vulcan system of gearing is employed, that is, Föttinger fluid couplings absorb the torque inequalities of the Diesel engines and any torsional oscillation which might manifest themselves in the system. In this way the gearing is protected and, it is claimed, its useful life extended. In the *Milwaukee*, on the other hand, the Frahm system is used, wherein the engines are rigidly coupled to the pinions in much the same way as some turbines are connected to their pinions in geared installations. By careful design, danger from torsional vibrations is avoided and the gearing teeth are able to operate satisfactorily over long periods. In fact, the performance of earlier geared motorships fitted with this type of drive, notably the *Monte Sarmiento*, *Monte Olivia* and *Monte Cervantes*, indicates that the useful life of gearing constructed on the Frahm system will be in no way inferior to that possible with any other arrangement, notwithstanding the fact that the engines are made directly reversible, while, in the case of these earlier ships, four-stroke cycle machinery is employed. In view of the undoubted advantages in respect to weight and cost reduction and economy of engine height, it is perhaps strange that the geared Diesel system makes no progress in this country. German engineers and shipowners have demonstrated that its reliability compares favorably with that of direct-coupled engines of the heavy slow-speed type, and it is to be hoped that the successful operation of later geared motorships, like the *St. Louis* and *Milwaukee*, will result in British owners giving the system their serious consideration and, better still, a trial in an actual vessel.

TRUNK-PISTON ENGINES.

One of the most interesting motorships to be completed during 1929 was the twin-screw Norwegian cargo carrier *Villanger*. This vessel is notable in being the first large British-built-and-engined motorship to be fitted with trunk-piston machinery. She is a fast vessel of about 8500 tons deadweight, and is owned by Westfal Larsen, of Bergen, a firm who have exploited the trunk-piston Diesel engine for comparatively full form cargo ships with conspicuous success. The *Villanger* is propelled by two Kincaid-built Burmeister & Wain type engines, each having six cylinders 590 millimeters in diameter by 1100 millimeters stroke. They are of the standard air-injection type and are capable of developing an aggregate power of 4000 I.H.P. at about 160 R.P.M. The use of trunk-pistons for cargo ship propulsion is another development which has met with considerable application on the Continent, but which, up to the present, has entirely failed to meet with the approval of British shipowners. Here again, cost and weight are economized without any sacrifice of reliability. A great many British engineers have a strong prejudice against the use of trunk-piston engines for mechanical reasons, but Scandinavian owners have, for several years, shown their entire practicability, durability and economy. Perhaps, when the Bel-

fast Steamship Company's cross-Channel motorships are all completed and have been in service for the space of a year or so, this subject will receive further ventilation in British marine engineering circles, for these ships are propelled by large four-stroke cycle airless-injection engines of the trunk-piston type. The power developed by these engines is considerably in excess of that usually met with in machinery of this class, and no doubt if they show up well in service—and we have not the slightest doubt but that they will do so—trunk-piston machinery for main propulsion may be seriously considered by British shipowners.

It will be recalled that after careful and lengthy experiments Sulzer Bros., Winterthur, have recently accepted an order for an eight-cylinder double-acting two-stroke cycle engine for installation in a vessel for the Nederland Company. This engine will develop 7000 B.H.P. at 100 R.P.M. (the same power and revolutions of the first B. & W. two-stroke cycle double-acting engine, in passing), but at the time of writing we are not in a position to give any technical details of the engine. These makers have steadfastly adhered to the single-acting two-stroke principle for many years with, of course, very great success. Doubtless, however, they feel that the time has now come to offer a double-acting engine, particularly for cargo ship work, for those shipowners to whom this type of power unit appeals. The appearance of a Sulzer double-acting engine design, however, in no way represents a "straw in the wind," for the firm are as strong as ever in their belief in the wide appeal and economic advantages of the two-stroke cycle single-acting engine, which, one gathers, will continue to be their principal type of marine oil engine.—"Shipbuilding and Shipping Record," January 1, 1930.

MARINE OIL-ENGINE PROGRESS IN 1929.

Airless-injection Engines, Exhaust-gas Pressure Charging, the Use of Waste-heat Boilers and the Employment of Trunk-piston Engines Form Notable Features of Last Year's Developments.

That there should be four entirely new types of Diesel motor placed on the market by important manufacturers last year, 17 years after the first large oil-engined ship went to sea, is an indication of the vitality in the industry. But these represent only a small part of the activity that was evinced in all directions, leading, in one form or another, to an increase in efficiency of motor ship machinery. In many respects the energy shown has been greater than for several years past, builders and designers unquestionably being influenced by the endeavors of steam-engine builders to make up some of the ground they have lost during the past ten years.

The total power of all the internal-combustion engines built last year for installation in ships is slightly in excess of 914,000 I.H.P.—considerably greater than the corresponding figure of 1928—but at the present time there are oil engines of 1,777,000 I.H.P. on order.

Despite the almost general predictions to the contrary, four-stroke single-acting motors are easily holding their own in competition with the two-stroke design. The details are given in the following table:—

CLASSES OF MACHINERY FITTED IN VESSELS BUILT IN 1928 AND 1929.

	1928.	1929.
Number of ships	189	181
Four-stroke single-acting	100	107
Four-stroke double-acting	4	8
Two-stroke single-acting	73	53
Two-stroke double-acting	12	13

The percentage of four-stroke engines was thus increased from 55 per cent in 1928 to 64 per cent in 1929.

The position in regard to motor vessels under construction, in order, is set forth below:—

MACHINERY TO BE INSTALLED IN MOTOR SHIPS ON ORDER.

Number of ships	351
Four-stroke single-acting	217
Four-stroke double-acting	6
Two-stroke single-acting	113
Two-stroke double-acting	15

The proportion of four-stroke engines of all types to be installed in ships now on order is, therefore, 62 per cent.

Whatever view one may take regarding the respective differences in classes of engine, it is hardly in doubt that the competition of the two-stroke motor has been largely met on account of the activity and initiative that have been shown by the manufacturers of four-stroke machinery, particularly in its adaptation in connection with exhaust-pressure charging, the general employment of airless injection of fuel, the use in many instances of trunk-piston machinery and the widespread employment of waste-heat boilers. It has, for instance, been found that in many installations not only is the price of four-stroke machinery lower than that of corresponding two-stroke plant, but the weight and space occupied, by adopting all the expedients possible, have been brought within close range of those of the competitive designs.

In making this comment, it is not intended to imply that the four-stroke engine will forge ahead in comparison with the two-stroke type, but that the competition between them is likely to be maintained on very acute lines indefinitely, so far as one can observe. In any case, what has happened during the past two years has disproved the belief that the two-stroke motor will supplant the four-stroke type within a measurable period.

ENGINE TYPES.

It is not without interest to note the actual types of engine which were installed in the vessels completed during the past year, and this is summarized in the following table:—

TYPES OF ENGINE COMPLETED IN 1929.

	No. of Ships.
Burmeister and Wain	
Single-acting	81
Double-acting	6

Sulzer		24
M.A.N.		
Double-acting	12	
Single-acting	6	
		18
Doxford		8
Werkspoor		7
Fiat		6
Other types		31
Total		181

Corresponding details of the machinery to be fitted in the vessels now on order are given in the table below:—

TYPES OF MACHINERY TO BE FITTED IN SHIPS ON ORDER.

	No. of Ships.	
Burmeister and Wain		
Single-acting	147	
Double-acting four-stroke	6	
Double-acting two-stroke	2	
		155
Sulzer		
Single-acting two-stroke	70	
Double-acting two-stroke	1	
		71
M.A.N.		
Double-acting two-stroke	12	
Single-acting two-stroke	13	
Single-acting four-stroke	18	
		43
Doxford		24
Tosi		
Single-acting two-stroke	4	
Single-acting four-stroke	6	
		10
Polar		10
Russian		9
Fiat		7
Werkspoor		6
Other types		16
Total		351

NEW ENGINES.

Two of the new engines which were brought out in 1929 were of the double-acting two-stroke type, one built by the Deutsche Werke and installed in two fast cargo ships, the *Sud Expreso* and *Sud Americano*, and the other, more interesting in every respect, developed by Burmeister and Wain and installed in the passenger and cargo ship *Amerika*, which is now completing.

The chief feature of the Deutsche Werke design and which alone distinguishes it from what might be termed standard double-acting two-stroke

practice, is the adoption of identical covers at the top and the bottom of the cylinders.* This is only rendered possible by having two horizontally disposed fuel valves at the top of the cylinder as well as at the bottom, while the starting-air valve is located in the center at the top, vertically, the hole for the starting-air valve cage being similar in size to that which takes the piston-rod gland at the bottom.

The Burmeister and Wain double-acting engine, on the other hand, breaks all rules of accepted practice. Only one engine has been built and, so far as is known, no other is under construction. It is thus too early to give any detailed opinion as to its further possibilities. The main novelty, and it is considerable, is the employment of exhaust valves, instead of ports in the center of the cylinder. The reason for their adoption is to enable a thorough scavenge from the scavenging ports in the center to the exhaust valves at the top and bottom. This, in turn, permits of the employment of higher mean effective pressures than with the exhaust-port system (100 pounds per square inch M.I.P.) and brings about a corresponding reduction in specific weight, which is stated to be about 120 pounds per B.H.P. for an engine operating at 100 R.P.M. Airless injection is employed and a fuel consumption of 0.38 pound per B.H.P.-hour is claimed.†

Before concluding the reference to double-acting two-stroke engines it remains to be recorded that towards the end of the year an order was placed for the first double-acting two-stroke Sulzer engine. This is a 7000 B.H.P. eight-cylinder unit.

AIRLESS-INJECTION ENGINES.

At the beginning of 1929 the airless-injection Diesel engines on order probably did not total more than 50,000 horsepower. Today the figure is nearer 500,000 horsepower, a simple fact which illustrates clearly enough the rapid progress that has been made within the past 12 months. It has already been mentioned that the new B. and W. double-acting two-stroke engine is of the airless-injection design and motors were placed under construction by the M.A.N. during the year also to operate on airless injection and of the double-acting two-stroke type. These will be the first marine units of their class.

Further, numerous examples of Burmeister and Wain single-acting motors with airless injection were placed in service during the year, and the first of the crosshead type so equipped were installed in the Blue Funnel cargo liner *Agamemnon*. Harland and Wolff constructed their first machinery of the class (of the trunk-piston type) for the *Ulster Monarch* and her sister ships, and there are now being built engines of 5500 B.H.P. in twelve cylinders, with trunk pistons, for the Pacific Steam Navigation Co., and a ten-cylinder unit for the Bergen S.S. Co.'s Newcastle-Bergen mail and passenger vessel. Orders were placed for a Spanish passenger ship to have M.A.N. four-stroke airless-injection machinery totalling about 7400 B.H.P., while the new Polar two-stroke airless-injection engine was developed; also a four-stroke type, the Winton, in America. The further adoption of airless-injection Diesel engines will undoubtedly be one of the main characteristics of internal-combustion machinery development in the next few years.

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* "The British Motor Ship," September, 1929, page 214, and October, 1929, page 255.
† "The British Motor Ship," October, 1929.

PRESSURE CHARGING.

Pressure charging gained remarkably in favor during the past year, although the first installations are only now coming into service. But already there are four competitive systems of pressure charging and probably the engines now in course of construction to be so fitted total approximately 300,000 horsepower. There is, therefore, no question of experiment and it is more than probable if the *Oceanic*, the 60,000-ton White Star liner, is Diesel driven, that pressure charging will be employed for the engines driving the dynamos.

Several of the applications were in outstanding vessels. They included the installation of the Blue Funnel cargo liner *Agamemnon* in which the normal output of 6000 S.H.P., with twin-screw four-stroke single-acting machinery, was raised to 8600 B.H.P. by the adoption of exhaust-gas charging with Rateau turbo-blowers; during the shop trials the machinery attained an output of 10,060 B.H.P. With the tanker *Megara* the speed of what would normally have been an 11-knot ship was increased to 12½ knots on service, the Werkspoor pressure-charging system being employed in this instance. The increase in power amounted to 50 per cent. It may be added that the mean indicated pressure in normal service is 118 pounds per square inch, and the importance of the system may be gauged from the fact that four ships of the class of the *Megara* can do the work of five which are unsupercharged.

The 12,000-ton liner *Llangibby Castle* was the first passenger ship to be pressure charged on the Büchi system, the four-cycle single-acting machinery developing a maximum of about 8600 B.H.P. The significance of pressure charging will be noted at once, namely, that in addition to other characteristics it enables ships requiring power up to 10,000 S.H.P. to be equipped with four-stroke single-acting machinery, and this had not hitherto been possible. A further instance will be afforded in the Bergen-Newcastle mail ship for the Bergen S.S. Co., in which 10,000 S.H.P. will be developed on twin-screws with four-stroke single-acting trunk-piston machinery of the airless-injection class.

EXHAUST-GAS BOILERS.

The rapidly increasing adoption of exhaust gas boilers may be noted at this point, for they were utilized for the first time in the liner *Llangibby Castle*, in conjunction with the pressure-charging system, while they have been applied also for the first time this year with two-stroke engines in the large tanker *Calgarolite*, fitted with Krupp engines, and the Doxford-engined *Sheaf Holme*. In the former, the Clarkson thimble-tube design was employed, and in the latter a new boiler developed by Cochrans with combined oil-firing and exhaust-gas firing. The results with all exhaust-gas boilers appear to be eminently satisfactory and they have afforded, in tankers and corresponding vessels of approximately 3000 B.H.P. machinery power, a saving of from 1½ to 2 tons of oil daily. The largest installation will be seen in the liner *Britannic*, now in course of construction and in which machinery of 20,000 S.H.P. is to be fitted. It may roughly be estimated that with four-stroke single-acting engines ¾ pound of steam can be raised per B.H.P., and with two-stroke engines up to ½ pound of steam per B.H.P. It may be added that some of the exhaust-gas boilers now used also act as silencers, so that no other silencing devices are needed.

TRUNK-PISTON ENGINES.

It has already been indicated that trunk-piston engines, far from losing favor, are gaining ground, and it is probable that 20 per cent of the ships turned out in 1929 were equipped with this class of machinery, and an equal proportion is now on order. The question of high lubricating oil consumption has been solved, and the figures recorded with the latest type of trunk-piston engine are approximately those of the crosshead design. It must be recorded, however, that in this country shipowners have shown little tendency to adopt the trunk-piston type compared with what has been accomplished in Scandinavia, but on the other hand the majority of motor ships building for Norwegian owners are equipped with this class of machinery.

Examples were, however, found in the Harland-B. and W.-engined *Ulster Monarch* and her sister ships, while the first Kincaid-B. and W. trunk-piston engines were constructed for installation, however, in three Norwegian-owned cargo vessels built at Armstrong, Whitworth and Co.'s yard. There is a 17,000-ton passenger liner for the P.S.N. Co. under construction by Harland and Wolff, in which quadruple-screw trunk-piston single-acting machinery of about 22,000 S.H.P. will be installed.

NEW TWO-STROKE ENGINES.

Two new two-stroke single-acting engines were developed during the course of the year, the first being the Krupp design, as fitted in the *Calgarolite* and notable for the employment of special cylinder covers, made of a special cast iron and designed so that the central portion is a simple and light casting with an opening to accommodate the fuel-valve housing. The piston head is of forged steel and the scavenging pumps, of which there are three, are equally distributed along the length of the engine and are attached directly to the cylinder blocks.*

The other new engine is a Tosi air-injection type, of which full details with illustrations are published on another page of this issue.

HIGH-POWERED ENGINES.

The high-powered marine Diesel engine does not lie exclusively in the domain of two-stroke double-acting machinery, for the biggest unit actually fitted on a ship during the course of the year was a single-acting two-stroke type of 7040 B.H.P. The two engines of the *Britannic*, however, were completed but not installed in the vessel, these being of 10,000 S.H.P. each and of the four-stroke double-acting design.

Even with four-stroke single-acting machinery the maximum outputs are rising, so that in vessels which have machinery up to 10,000 B.H.P. this type of motor can now compete. The *Agamemnon* was equipped with plant capable of this output, when supercharged to the maximum, and, as recorded above, several sets of 5000 B.H.P. are being built. When it is considered that in the *Agamemnon* an engine-room no more than 55 feet in length encloses machinery of from 9000 S.H.P. to 10,000 S.H.P. of the four-stroke single-acting type, it is clear that such plant suffers comparatively little disadvantage in the matter of size—to such an extent has the employment of the expedients mentioned above helped the four-stroke single-acting engine. At the same time it must be admitted that pressure charging continuously, with

* "The British Motor Ship," June, 1929.

the object of increasing power 50 per cent, has yet to be confirmed in its suitability over long periods, although the results achieved by the *Megara*, the *Lochmonar* and other vessels seem to leave no doubt on this score.

GEARING AND ELECTRIC DRIVE.

The most important application of the geared Diesel drive was made last year in the *Milwaukee* and *St. Louis*, two intermediate 16,500-ton transatlantic passenger liners for the Hamburg-Amerika Line, and so far as can be gathered the arrangement has proved successful. It was of peculiar interest in the combination of double-acting two-stroke engines running at 215 R.P.M., with the geared drive, giving a propeller speed of 110 R.P.M. The weight of the engines and gearing is understood to be in the neighborhood of 125 pounds per B.H.P., and the low height is naturally advantageous in a passenger liner.

It is, however, somewhat difficult entirely to eliminate noise in geared Diesel ships, and perhaps the application of the system will find wider scope in cargo ships. It is to be utilized in a 9500-ton vessel of this class for the Rotterdam Lloyd, with two double-acting two-stroke engines of 2750 B.H.P., and the vessel will be run on the same service as a number of single-acting two-stroke single-screw ships of precisely similar dimensions and speed.

Diesel electric propulsion, on the other hand, has made no progress, and in Europe not a single vessel with this method of drive was constructed. Two were brought into service in America, being conversions from steam, but the cost appears to have been high.

THE ADVANCE OF STANDARDIZATION.

While there has been no trend in the least towards the unification of types of Diesel engines, there has been a very marked movement in the direction of the standardization of individual design, leading to a reduction in cost and a rapidity of output that is not otherwise possible. This is specially notable in the orders that have been placed within the past six months, and on the order books of builders such as Harland and Wolff, J. G. Kincaid and Co., William Doxford and Sons, the Wallsend Slipway and Engineering Co., will be found contracts for batches of between six and a dozen six or eight-cylinder engines precisely similar in every respect. On the Continent the situation is even more remarkable, and probably in the case of Götaverken, Burmeister and Wain, Sulzer Bros., and the Eriksbergs Mek. Verk., there are batches of motors of 20 or more standardized in every particular. This development is advantageous, both to the shipowner and the engine manufacturer, and has no disadvantage, while the variation of power that can be obtained with a standard engine by increasing or decreasing the number of cylinders affords a range which enables a shipowner to adopt a standard engine with almost any class of ship which he desires to build.

AUXILIARY PLANT.

One or two tendencies appear to be growing more definite in relation to auxiliary plant for motor ships. In the first place, somewhat larger power is now being provided, while a higher speed engine is being commonly utilized (up to 400 R.P.M.), with a reduction in weight and space. The airless-injection engine is making very rapid progress, and, finally, there is a certain departure in several cases in that shipbuilders and propelling-engine machinery manufacturers are adopting, in many instances, the policy of

steam-engine builders of ordering their auxiliary machinery from specialists in the construction of this class of plant. There is a good deal to be said for such a development, since the quantity production of engines up to 400 B.H.P. or 500 B.H.P. involves an entirely different technique from that of the construction of a more limited number of propelling motors varying from 1000 to 10,000 B.H.P.

FUEL CONSUMPTION.

Although there has been no revolutionary change, it can definitely be stated that the fuel consumption of motor ships with the newer classes of machinery and the latest developments in auxiliary plant shows signs of diminishing. Figures were published during the year showing an overall consumption of no more than 0.365 pound per B.H.P.-hour for a Doxford-engined ship, while nearly all the four-stroke airless-injection engines of the new type and many of the two-stroke design have guaranteed consumptions not exceeding 0.38 pound per B.H.P.-hour, the actual figure being probably lower. The utilization of exhaust-gas boilers, as stated, represents a further saving, often equivalent to 5 per cent, and it may be fairly stated that it is now possible to order a motor vessel with a fuel consumption 10 per cent lower than would have been feasible two years ago.

FURTHER PROGRESS.

Further progress will probably be upon the lines of steady improvements rather than revolutionary change. Airless injection will undoubtedly become more common, even for double-acting two-stroke units, the use of exhaust-gas boilers will be almost universal, and with four-stroke single-acting engines pressure charging will, no doubt, evolve rapidly. There are further fields open to the designer and attempts will have to be made to render the employment of lower-grade Diesel fuel satisfactory in Diesel machinery, should the price difference between Diesel oil and fuel oil continue in the United States. With the improvement in material it is possible that mean pressures in Diesel machinery will now be somewhat raised, so that the power-to-weight output will be increased, reflecting favorably on costs of construction. The adoption of engines of higher speed than previously, especially in relatively fast vessels, is likely to be a development of 1930, and motors turning at 300 R.P.M., driving propellers in conjunction with mechanical gearing and hydraulic clutches, may receive more attention than in the past. The year 1930 opens with the Diesel-engine industry in a more prosperous condition than ever before.—"The British Motorship," January, 1930.

THE PROPULSION OF FAST TRANSATLANTIC LINERS.*

By Professor Dr. G. Bauer.

In considering the latest development in the design of fast liners, as typified by the *Bremen*, which has considerably exceeded the guaranteed average voyage speed of 26¼ knots (during the first four voyages she maintained an average speed of 27.24 knots), the question arises whether the present state of technical development will allow this performance to be surpassed. This

* Abstract of paper read at the thirtieth meeting of the Schiffbautechnische Gesellschaft, held in Berlin on the 22nd and 23rd November, 1929.

may be answered in the affirmative, as it is not impossible on technical grounds to build transatlantic liners of higher speed; and the competition of the airship will stimulate the desire for faster ocean liners. The gain in speed may be secured either by increasing the engine performance or by reducing the size of the vessel.

The propelling installation of the *Bremen*, which presents a basis for further development, includes four boiler-rooms with eleven double-ended and nine single-ended water-tube boilers having 217 oil-burners and a total heating surface of 184,000 square feet. The steam pressure is 328 pounds per square inch, and the steam temperature 680 degrees F. Four sets of turbines, each consisting of three turbines working at 1800 R.P.M., drive the four screw propellers at a speed of 183 R.P.M. Each propeller has a diameter of 16 feet 5 inches, and has been designed to transmit 22,500 B.H.P. The daily oil consumption has been guaranteed not to exceed 806 tons, including that required for the Diesel generators. For the first four voyages the power required to drive the ship at 27.24 knots averaged from 100,000 to 105,000 B.H.P. The weight of the machinery, with water in the boilers and pipes, is 7350 tons, so that the specific weight per B.H.P. is 150 to 166 pounds. It is of interest to note that the boilers are not fitted with normal artificial draught, but that the closed boiler-rooms are under air pressure. The air is forced by fans into the boiler-rooms and flows through the air preheaters into the furnaces. This arrangement has the advantages that no air ducts are necessary, that the air pressure can be regulated easily, that the temperature in the boiler-rooms remains low, and that the boiler-rooms are very well ventilated. The fans have been made sufficiently large to lead the surplus air between the funnels and their casings, in order that the ship's accommodation should not be affected by the heat from the funnels. Certain of the engine-room auxiliaries are driven by steam turbines. For driving the circulating pumps and the air fans for the boiler-rooms, geared turbines are installed, while the oil pumps for the main turbines and gears and the main feed pumps are directly driven by turbines. All these auxiliaries are economical, require little space, and are not heavy.

If the *Bremen* is made the basis for further proposals for the propulsion of fast liners, it must be borne in mind that the plant in that vessel is operated with a small load only. The service air pressure does not exceed 1.4 to 1.6 inches of water, which might be increased without risk to 2½ or 3 inches. The fuel consumed by the boilers is a maximum of 0.45 pound of oil per square foot of heating surface, while in the German Navy as much as 1.02 to 1.22 pounds is allowed for such units. The turbine blades have been designed with a factor of safety of six, and the torsional deflection at the circumference of the pinions is less than 0.0004 inch. The maximum bending deflection is less than 0.00013 inch, and the load on the pinion teeth is about 730 pounds per inch of breadth. The load on all other parts is kept within similar low limits, so that the plant without any risk may develop 130,000 B.H.P., which would give the ship a speed of 28.5 knots. In favorable weather conditions the *Bremen* would always be able to maintain an average speed of 28¼ knots if her owners desired her to do so. She is well able to take the fuel required for this purpose. An increase of speed of one knot, however, would result in a reduction of the time required for the passage across the Atlantic of only four hours, and this would not be sufficient to bring the vessel into the category of a four-day ship.

A ship which has to cover the distance from Cherbourg to New York and *vice versa* within four days has 101 hours available for the westward run, corresponding to a speed of 31.3 knots, and 91 hours for the eastward trip,

corresponding to a speed of 34.8 knots. The difference arises from the variation in time between New York and Cherbourg. It is possible, however, to secure the advantage of almost a four-day passage even with a 100-hour return trip. The vessel could leave New York in the afternoon or at night, and arrive at Cherbourg in the early morning. This could be done by a vessel having a service speed of 31.5 knots and a trial speed of 32.5 knots.

Considering the technical and economical aspects of such a problem, the smallest dimensions which would accommodate the necessary machinery should be adopted, and this might be achieved with a hull having a length of 820 to 850 feet, a beam of 88 to 92 feet, and a draught of 26 to 30 feet, with a displacement of from 31,000 to 33,000 tons. Such a vessel might be satisfactorily driven by three shafts of 50,000 B.H.P. each, with four turbines to each shaft having a speed of 2000 R.P.M., reduced by gearing to 200 R.P.M. The boiler plant might be similar to that of the *Bremen*, with a heating surface of about 175,000 square feet, which, of course, would be loaded more severely. The pressure would have to be fixed at 430 pounds per square inch, and the steam temperature at 750 degrees F. The weight of this plant would amount to 8100 tons, including the water in the boilers and the lubricating oil. The bunkers would have a capacity of 5400 tons. If the ship were designed with four shafts, the general arrangement of the machinery would be similar, but the speeds of the turbines and propellers would be 2300 and 230 R.P.M., respectively, and the weight 8350 tons. There is no doubt that the four-shaft arrangement is to be preferred at present—although it is a little more complicated—as the units are smaller and the plant more reliable.

The steam-pressure limit must be chosen with due regard to economy, and also to the difficulties which arise from too great an increase. The economy which can be gained by a pressure increase from 328 to 542 pounds per square inch is, in fact, only 5.1 per cent if the steam temperature is 680 degrees F. The same gain is obtained if the pressure is kept at 328 pounds per square inch with an increased temperature of 790 degrees F. If both measures are applied, the gain is 10.1 per cent. From a pressure of 328 pounds per square inch with 790 degrees F. steam temperature to 1430 pounds per square inch and 790 degrees F., there is a gain in economy of 10.5 per cent. All these figures are calculated on the basis of a feed-water temperature of 236 degrees F. It must be accepted, however, that within the next two or three years, pressures of more than 600 to 700 pounds per square inch will not be seriously considered for marine purposes, for reasons of reliability.

A design of a four-day transatlatic liner with direct-acting Diesel engines cannot be considered at present, as the risk of building such large oil engines would be too great. The indirect drive might be achieved with the aid of Vulcan geared hydraulic couplings, and by utilizing four high-speed, double-acting, two-stroke cycle, airless-injection, Diesel engines of 6850 B.H.P. on each side line of shafting, and with six such engines on the two middle lines of shafting. The engines would run at a speed of 280 R.P.M., and the shafts at 210 R.P.M.

The electric drive has been considered for such vessels, but a thorough investigation has shown that it cannot be considered to be as favorable as the foregoing designs. The transmission losses with electric drive are asserted to amount to 5 per cent in very large plants, but this figure has certainly not been attained in practical service. The losses in gearing are not larger than 1 or 2 per cent, and the losses due to the astern turbine amount to 1 to 1½ per cent. It is stated by the advocates of electric drive

that the turbines employed with it can be designed on the lines of those for stationary plants, with increased economy arising from higher speeds of revolution and smaller blade clearances; but the use of such turbines cannot be considered for marine drive, as they have not proved to be sufficiently reliable. If redesigned for application on board ship, such turbines would lose a good number of their advantages. The specific fuel consumption of the *Bremen* is more than 10 per cent lower than that of the electrically-driven *California*. Again, the electric drive is more complex; it is heavier, in spite of the progress made in electrical construction; and the cost of electrically-driven ships is at least about 20 per cent greater—an increase which is very important in the case of a modern fast liner. Electric-transmission plants are very complicated internally, and there arises the danger that the mechanical engineers who would have to operate them on board ship would not be able to handle them satisfactorily in all cases or to undertake their repair if anything went wrong. The acknowledged superiority in economy of electric plants for cruising speeds is of no importance for transatlantic liners. The prospects of the Diesel-electric drive are still worse than those of the turbo-electric drive, as the only gain by the adoption of the former in comparison with Vulcan-geared Diesels would be that these gears—which are absolutely reliable and noiseless—would be dispensed with.

Transatlantic shipping will have a competitor in the airship, the possibilities of which have been proved by the performances last summer of the German lighter-than-air craft. Airships, however, are greatly influenced by wind and weather, especially over the Atlantic, and the shortening of the passage by virtue of their higher speeds is not of such great importance. The conditions are much more favorable for the airship on the routes to the Far East and to Australia, as the passage from Berlin to Japan may be cut down from 45 days with the steamship to 4 or 5 days with the airship. It is doubtful whether it will be possible in the near future to construct airships to cross the Atlantic with the reliability of the fast steamship, which at least affords some comfort to the passengers. The problem of the transatlantic express liner, therefore, will not be obsolete in the near future.—"The Shipbuilder," December, 1929.

WELDED BOILERS.*

By E. R. Fish.†

The idea of fabricating high pressure steam boilers by welding processes is not new. It has, however, been practiced to a considerably greater extent abroad than in the United States. There have been some serious failures of welded boilers that have to a very great degree discouraged its use here. So far as the writer is aware, there are no high pressure boilers constructed by the fusion welding process in this country, but there are some forge welded ones. Therefore, so far as the adaptation of welding processes to actual boilers is concerned, there is but little to discuss. You will note that reference is made to high pressure boilers, for the above statements refer only to boilers which are designed to carry pressures in excess of fifteen pounds per square inch.

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Low pressure boilers used practically exclusively for heating purposes are manufactured on a production scale in tremendous volume without the use of any riveting whatever. This, of course, involves only relatively light plate, rarely over 7/16 inch thick. The A. S. M. E. Boiler Code for Low Pressure Heating Boilers specifically recognizes the progress of the art by permitting this method of joining low carbon steel plates in fabricating boilers of this class. There are literally thousands of welded heating boilers in use with a very gratifyingly low record of failures. But it is with power boilers that we are now particularly concerned. First let us examine the hazards involved. As the pressure increases the boiling point of water rises. If the pressure be reduced, the stored heat immediately evaporates a portion of the water and if the reduction of pressure is sudden there is created an explosive action similar to that of gunpowder. Professor R. H. Thurston some years ago calculated the stored energy in boilers which very clearly illustrates why boiler failures are so disastrous.

This energy is divided between the steam and water. For each pound in weight, that in the steam is very much greater, at the usual operating pressures than in the water, but as the number of pounds of steam in a boiler is much less than the weight of water, the total energy in the latter is far greater than that in the steam and hence has the greatest destructive potentialities. To give an idea of these quantities, the following table has been prepared. These figures are approximate only and are not given as being exact.

	When Reduced to Atmospheric Pressure and 212 Deg. Fahr.			
Pres. by Gauge	Available Energy in Ft. Lb. In. One Lb. Water	One Lb. Steam	30,000 Lb.* Water	90 Cu. Ft.* Steam
5	146	16,800	4,380,000	57,500
25	1,600	54,000	48,000,000	463,000
50	3,875	78,800	116,250,000	1,064,000
75	6,050	94,800	181,500,000	1,740,000
100	8,100	106,000	243,000,000	2,460,000
200	15,000	130,000	450,000,000	5,475,000
500	30,100	166,000	903,000,000	16,400,000
1,000	49,000	180,000	1,470,000,000	35,650,000

One of Dr. Thurston's concrete examples is of a watertube boiler of 3000 square feet of heating surface, ordinarily referred to as a 300 horsepower size, and at 100 pounds working pressure, a very low one today, in which there would be available stored energy in the water and steam of nearly 110,000,000 foot-pounds. If suddenly released and all applied to moving the boiler structure, this energy could project the boiler to a height of over 2000 feet.

It will be evident therefore that the destructive potentialities of high pressures are very much greater than at the lower pressures. It is because of this consideration of disastrous possibilities alone, that has deterred those in authority from heretofore accepting any except forge welding in the construction of high pressure boilers. Records are replete with failures of autogenously welded pressure containers, the reasons for which may be attributed to the relative inexpensiveness and ease with which anyone can

* Typical 500-horsepower water tube boiler.

set up a welding shop, and through the rapid expansion of the use of welding in fabricating containers and pressure vessels of many kinds. This has resulted in the establishment of many shops to fabricate vessels of this sort, frequently without proper precautions as to the competency of the welders and methods and often without an appreciation of the responsibilities involved. In addition, there has been the necessity of learning by experience, by even the most conscientious and painstaking fabricators how to solve the problems that developed. It is now recognized that sufficient progress has been made to justify serious consideration of the application of this kind of welding as it relates to boilers. But it may be asked, why the necessity for departing from the long accepted and practiced method of joining plates by tried and true riveting? There are two reasons.

(1) The rapid development of power plant practice, particularly the adoption of the steam turbine, made unforeseen demands on the boiler makers. Unheard of volumes of steam per hour were insistently demanded. When reciprocating slow speed engines were the prime movers, the engine room was the dominating part of a power plant as regards space. Pressures and steam temperatures were also limited. But the turbine entirely upset the situation. Requiring only possibly one-tenth the space needed by the old engines of the same capacity, the boiler room became dominant as to space, and efforts were directed to getting more and more steam from single units both by increasing the steaming capacity and the size of boilers. Had that been the limit of the demand it would not have been so hard on the boiler manufacturer, but the economy of and consequent demand for higher and higher pressures very greatly complicated the manufacturing problems which had to be faced. It is the cylindrical drums that constitute the greatest difficulty. Practical considerations limit the thickness of plate which can be satisfactorily riveted to about 2½ inches, and there are but few shops that can go that high. Couple this with shell lengths up to 40 feet or more and it becomes evident such structures are not easily made. For riveted construction, the usual diameters of drums therefore limit possible pressures to the order of 600 pounds per square inch, which, however, is now not regarded as very high. There are a dozen or more large units running today at 1200 pounds to 1400 pounds and three in process of installation to carry 1840 pounds. The thickness of shells for such pressures is over 5 inches, quite beyond any possibility of riveting. Up to the present all such are seamless, forged from a single ingot with the heads integral. However, high pressure chambers for oil refineries with wall thicknesses up to 5 or 6 inches have been fabricated by fusion welding. All such are made of several plates formed to shape hot and rigidly held in place while being worked on. There are a great number of pressure stills of this type in use and the reported failures are very few. This is excellent evidence of the sufficiency of good welding.

(2) The mysterious cracking of plates of boiler drums, at first in some isolated and fairly well-defined localities and later much more widely distributed, led to investigations that indicated quite positively that certain water characteristics are the principal causes of this cracking which always occurs where plates, rivets or tubes are in close contact. This theory has become generally known as caustic embrittlement and will be referred to hereafter as plate embrittlement.

It has been the thought of some engineers that the elimination of riveted construction would avoid trouble of this kind. However, it can be greatly minimized, if indeed not entirely avoided, by inside calking as is now quite generally practiced. The need for welding instead of riveting in order to prevent possible embrittlement, in the plate thicknesses that admit of either

method of construction, is therefore not necessarily demanded although some prefer not to take any chances whatever. The possibility of this phenomenon appearing in all types of fabrication is always present at the tube holes, since the method of inserting and rolling tubes is identical for all.

The applications of welding to boiler fabrication may be divided into two principal classes with a possible third and fourth minor ones—forge or hammer welding, autogenous or fusion welding, electric resistance welding and thermit. The former is hoary with age, having been used since the beginning of the art of the working of iron and steel. However, the problems involved in welding the edges of long plates is quite a different matter from that of limited areas. But the reasonable certainty with which the edges of plates may be joined by this process caused it to be permitted although with restrictions. The quality of the steel best suited to this purpose is such as to give physical characteristics less than the regular flange or fire-box qualities. To be readily weldable the carbon content must not exceed .15 per cent and in consequence the tensile strength runs from 45,000 pounds to 55,000 pounds as against a range of 55,000 pounds-65,000 pounds for fire-box quality. The maximum thickness of plate that can be forge welded is about 2 inches and present facilities limit the length to 30 feet. It will be seen therefore that this method, as applied to drums is limited to pressures of about 500 pounds. It is also used to a considerable extent in the manufacture of headers for watertube boilers, but that is quite a specialized branch. Recognizing that there is some uncertainty as to the integrity of the weld the unit stress to be used in making calculations is limited to 7000 pounds as against 11,000 pounds for riveted joints and flange or fire-box quality of plate.

The second welding process is the autogenous, either electric or oxy-acetylene, wherein extraneous metal is introduced to affect the bond between the edges. It will be appreciated that the thicknesses concerned are such that the weld metal must be deposited by a series of operations and that its mass is considerable. In the course of its operation a boiler is subjected to many unmeasurable stresses and distortions of shape due to pressure and temperature changes. It is because of these that ductile material is so essential. It is imperative that the weld metal shall partake to a reasonable degree of the physical characteristics of the base metal. Only in this way can freedom from the effect of the localization of the unknown stresses be assured. Until very recently the character of the deposited metal has been uncertain and variable. Improvements in both methods and material now make it possible to meet the requirements. In order to insure uniformity in such large welds, automatic welding machines are highly desirable although not absolutely necessary. Fortunately it is not likely that there will be defects at the same place in each of the layers, so the risk of a poor weld is minimized.

The third method which is of very limited scope is electric resistance welding, which has been recognized as the equivalent of forge welding. It is as yet applicable only to the joining of very small surfaces and hence is not included as one of the methods for fabricating large drums or other parts.

The fourth method of welding the edges of plates is by the use of thermit. This has been talked of but experiments of only the most primary character have as yet been made so the idea is in the preliminary stage. To it will doubtless apply the same restrictions as affect the fusion method. It has the advantage that the nature of the deposited metal is more readily controlled than is the case with the others. The cost, however, would probably be against it and outweigh the advantages. It is mentioned here as a possibility.

The local heating of large plates inevitably sets up stresses in the colder parts contiguous to the welds and which must be relieved before being put to use. It is therefore necessary to "stress relieve" the stucture when the welding is completed. From the dimensions heretofore given it is evident that a very large heating furnace will be a necessary part of the equipment for any shop which may undertake to make welded drums by whatever method.

How can the soundness of a welded vessel be assured is an inevitable question. It has been asked innumerable times and a satisfactory answer must be given, but about this there are wide differences of opinion. It is, of course, manifestly impossible to investigate every weld by cutting at frequent intervals or by testing to destruction. A non-destructive test that will prove the integrity of the weld with reasonable certainty is required. It has been proposed and practiced to some extent to vigorously hammer the weld or the immediate vicinity while under a test pressure of twice or more the designed working pressure. Exception has been taken to this practice. There have recently appeared two suggestions for other methods. One is a modification of the Sperry scheme for detecting defective rails. There follows extracts from a paper by Mr. Sperry.

"With the successful application of the electric method of inspection to the examination of rails in track it was but natural that the inspection of welds should be one of the first problems to be considered. The inspection of welding includes not only the location of pockets, cracks, inclusions, overlaps, etc., but also the detection of unwelded surfaces which frequently are held in intimate contact under heavy pressure produced by shrinkage in cooling the weld. The following research item had as its object the investigation of this phase of the problem."

"The faces of two 1-inch steel cubes were so accurately ground, lapped and polished that they fit perfectly over the entire surface, it being possible to lift one cube by its well-known adhesion to the other. The polished surfaces were then carefully washed with ether to remove any trace of grease or dirt. The cubes were now pressed together between two copper outer terminal pads with a pressure of 5000 pounds. A current of 100 ampères was passed through the blocks and the average voltage drop noted over unit lengths of the sound metal at different points, then over the same unit containing the junction of the two blocks. Here were two surfaces as nearly perfect as could be made, pressed together with 5000 pounds, yet the electrical resistance of the unit length containing the gap was one hundred seventy-five times the resistance of an equal length of sound metal. This demonstrates that the electrical method is applicable to the detection of any interruption of the molecular continuity of a metal. Many fissures are practically sub-microscopic.

"The initial work on the testing of welds by the electric method was performed on welds joining one-half-inch cold rolled plates. Both gas and electric welds were tried, each sample containing a hidden defective spot. An adaptation of the standard test method was quickly developed, which made possible the accurate location of the defective spots. This method is not only applicable to welded plates but may be applied to the inspection of welded drums, pipes, etc.

"A plant has been constructed for testing welded joints of plates up to between three and four inches in thickness and forming drums of different diameters. The inspection to date is found to be accurate and dependable. It goes forward at a rapid pace and gives an ink record which is in the nature of a certificate of complete soundness of the weld. This apparatus

is capable of adjustment so as to record very minute defects or it will record only those exceeding a certain predetermined area or percentage of a unit length of weld equal to the plate thickness. This widens out the adaptability and usefulness of the method. The circumferential seams are serviced without difficulty, sometimes more easily even than those running longitudinally as they require less length of primary conductor and less flexibility, but this is a minor detail.

"It would thus seem that the time is close at hand, if it has not already arrived, when by a simple combination of a well-known group of scientific facts and elements we are able to ascertain the internal structure throughout any weld or any critical part thereof, thereby finally establishing its complete integrity and freedom from hidden flaws of any type, establishing it as a 100 per cent union throughout its entire extent longitudinally.

"It is believed that electrical inspection should follow closely the welding process, especially on structures, so that if a hidden defect is discovered it can at once be dug out and corrected on the spot, reinspected and the process repeated until a 100 per cent weld is secured. It is hoped that the great ease with which the complete soundness of welds may quickly be tested and a permanent record secured by so simple and straightforward a method will give welding a new impetus, and by removing uncertainties will broaden its scope of application and usefulness."

The other method is based on sound transmission by the metal, using an ordinary physician's stethoscope and the X-ray, developed by Kinzel, Burgess and Lytle of the Union Carbon & Carbide Research Laboratory.

"In searching for a non-destructive test of welds the use of sound transmission or sound characteristics of the joint as affected by discontinuities and irregularities was investigated. It is known that the presence of a serious defect in a vessel can often be determined by its ring. The village blacksmith used this principle when he hammered metal cold on the anvil to test it by ring. In sounding a tank, however, there are a great many difficulties to be encountered, due to the forced and natural vibrations of the tank and the tendency for the natural vibration to drown out all other sounds. The authors have used the stethoscope to overcome these difficulties. The instrument consists of the ordinary physician's apparatus, with a gum rubber tip to exclude extraneous sounds and give contact on the irregular surface of the plate, as well as to minimize dampening of the oscillations at the contact of stethoscope and metal. The drum, pipe or plate in question is struck with a small hammer at the spot to be tested. The weight of the hammer and force of the blow depend upon the thickness of the plate, and the stethoscope is applied to the tank in the general vicinity of the striking zone. The sound heard at the instant of first tapping the joint is critical and is a function of the character of the material at the spot struck. Shortly after the tapping the waves are reflected and the sound picked up by the stethoscope becomes a composite of the natural vibration of the tank as a whole and of the forced vibration of the material immediately under the striking hammer. When the defect is very serious, however, it has a dampening effect on the total sound, so that its presence is also clearly evident in the after ring. The problem is analogous to that encountered in the well established method of testing staybolts by noting the character of the sound on striking. Testing staybolts in this way has proved to be very satisfactory.

"Each welded structure has a characteristic sound. After this is determined by brief preliminary tests the striking hammer and stethoscope are moved along the weld so that any irregularities are discernible."

The nature of the defects located by the stethoscope is of decided interest. Lack of fusion, particularly at the bottom of a vee or along the scarf gives

a characteristic reedy, high pitch, initial note. In many instances the points at which welding was started and stopped and where tacks have been made are also discernible with the stethoscope, but the characteristic tone is such that these points are readily different from lack of fusion. To date the authors have been able to check each other in most satisfactory manner in the testing of welded plates, pipes, pressure vessels and welded structural joints.

X-RAY EXAMINATION.

This method may be used either in conjunction with the stethoscope or as an independent sampling method. In the past most of the X-ray work on welds consisted in taking radiographic pictures through the weld, the X-rays passing in a line normal to the plane of the plate. As a result some idea of the porosity is noted, but the lack of fusion on the scarf in vee welds is not clearly registered on the film. By taking two photographs, one with the X-rays passing parallel to one scarf and one with the X-rays parallel to the other scarf, in addition to the photograph normal to the weld, this difficulty is eliminated. Thus, if there is lack of fusion with or without thorough penetration at the bottom of the vee or along the face of the scarf, X-rays pass along this place unabsorbed and leave their characteristic dark mark on the developed film. From the appearance of the films and a knowledge of the plate metal and welding rod used the ultimate strength of the weld may be estimated with considerable accuracy after a little experience.

A number of welded single V and double V coupons were X-rayed in the manner described, and the ultimate strength estimated from the films. In order to have some poor welds in the series a number were taken from the work of welders who had failed to pass the test required to qualify them for important work under procedure control. The authors had no knowledge of the condition of any particular weld prior to the X-raying. The coupons were then pulled in the tensile testing machine and the actual strength and the estimated strength compared.

The concordance of estimated and actual strength of the coupons tested is remarkable, and a maximum deviation of 3000 pounds per square inch resulted. In a 50,000 pounds per square inch weld, this is within 6 per cent of the actual value.

The general plan for a complete method of non-destructive testing of pressure vessels consists in going over the welded joints with the stethoscope and then making X-ray examinations of any points in question.

CONCLUSIONS.

1. Defects in line welds, girth seams or structural joints may be detected by the use of the stethoscope.

2. A quantitative estimate, good within 3000 pounds per square inch of the strength of the weld, can be obtained from X-ray photographs made in accordance with the above-described method.

3. A combination of the use of the stethoscope and X-ray constitutes a satisfactory quantitative non-destructive method of testing welds.

The complete papers describing these methods were presented at the recent fall meeting of the American Welding Society and appear in the September, 1929, number of its journal. These developments are cited to show that some of the uncertainties as to the soundness of welds can be definitely removed and that the problem is well along toward solution. Assurance that a weld is perfect will go far in furthering the willingness of insurance companies and boiler users to accept this method of construction. Both of the methods referred to, as well as the hammer test, are practicable shop operations.

If fusion welded drums are permitted as is quite likely in the not distant future, it will inevitably be under drastic restrictions that will prevent any but adequately equipped shops, both as regards men, methods and machines, from attempting to manufacture by this process. The tentative draft of the regulations now under consideration by the A. S. M. E. Code Committee is of interest. The following are the principal points:

Test plates from which specimens may be cut will have to be welded at the same time and under the same conditions as the shell.

Tension tests must be made which should fail in the plate, but if failure occurs in the weld the stress must not be less than the mean of the range for the plate.

Bend test specimens transverse of the weld must bend cold under free bending conditions until the stretch on the surface is 30 per cent with width of surface cracks deducted, none of which shall be longer than 10 per cent of the width of the specimen.

Impact test specimens from the bottom, middle and top shall have a minimum Charpy value of 20 feet pound.

Chemical analysis of weld metal with maximum requirements for manganese, phosphorus, sulphur and iron nitride.

The complete welded structure shall be heated uniformly to certain specified temperatures for different thicknesses, but averaging about 1100 degrees F. and held for one hour per inch of thickness for the purpose of relieving stresses due to welding.

The welded joints shall be explored by means of some form of nondestructive apparatus as will determine quantitatively the size of a defect.

After being stress relieved the vessel shall be subjected to 10,000 applications of hydrostatic pressure varying in a cycle from zero pounds to one and one-half times the working pressure, after which the welds shall again be explored by the nondestructive test apparatus.

Make macro and micro-photographs to show freedom from excessive porosity, incomplete fusion, laps, etc.

No holes shall be located in a weld nor shall a hole be nearer to the edge of the weld than the thickness of the plate with a minimum of 1 inch for plates less than 1 inch thick and 2 inches for plates of that thickness.

When constructed in accordance with the proposed specification the maximum unit working stress may be taken as 16 per cent of the minimum of the specified range for the plate.

It must be understood that these provisions have not been adopted and many of them may be changed or omitted and still others may be added before a final conclusion is reached. It is even possible that there may be enough objection raised so that a code acceptable to all the varied interests concerned cannot at this time be agreed to and that the matter will therefore be more or less indefinitely delayed.

It will be evident from the foregoing that at present, with the exception of forge welding, high pressure boilers may not be made otherwise than riveted or seamless, but that progress is being made and that probably fusion welding will soon receive recognition in the boiler field.

While the A. S. M. E. code does not now sanction the use of fusion welding subject to major stresses, revisions have been made that permit its use in many minor ways, particularly in small firebox types of boilers. In these the bottoms of waterlegs, fire door openings, furnace sheets, etc., may be welded provided the surfaces are adequately supported otherwise. Also attachments of various kinds can be affixed by welding, thus avoiding riveting that not infrequently causes annoyance by leaking besides being cheaper and really better. It will then be seen that the Code Committee has been

relenting in its attitude toward welding and is apparently on the verge of a major leap.

At about the point where it is no longer possible to use riveted construction it becomes practicable to use seamless forged drums, the most ideal structure for the purpose that can be imagined but very expensive. We have therefore welded construction in opposition to riveted and seamless for the whole range of thicknesses and the choice will be determined to a great extent if not entirely on the relative costs.

Where conditions are such as not to make welded construction either forged or fusion of paramount importance and first cost is a prime consideration riveted work will continue to be used. Roughly the former costs from two to three and one-half times that of the latter. That this disparity may in time be lessened is probable, but just now it is a real factor in making a decision.

The progress of the welding art is so rapid that any developments of the past can be of value only historically. The attempt has been made herein to deal only with the immediate present and imminent future without rehearsing the steps either at home or abroad of the ladder of experiences on which present knowledge and practice has mounted to its present status. It is hoped that what has been said will be of some interest and service to all.

METAL SPRAYING BY COMPRESSED AIR.

Many processes have been introduced for covering the surfaces of metals and other materials with a protective coat of metal, but the process in which the metal is sprayed by a jet of compressed air is one which possesses many points of interest to shipbuilders and marine engineers. This system of metallization, which is being developed in this country by the Metals Coating Co. Ltd., 22, Birchin Lane, London, E.C.3, can be employed for adding a wear-resisting surface to a softer material and for building up worn or pitted surfaces with the same metal as that of the worn or pitted part. For spraying the metal what may be regarded as a pistol is employed, to which is supplied, by means of flexible rubber hoses, oxygen and acetylene for the flame which melts the metal to be sprayed, and compressed air, which blows the molten metal in a finely atomized condition through the nozzle of the pistol on to the surface upon which it is to be deposited. The pistol also contains a small turbine operated by compressed air, which imparts a steady feed to the wire of metal or alloy which is being sprayed. The wire is fed into the hot zone of the oxy-acetylene blowpipe flame, where as quickly as it becomes molten, it is atomized by the air and projected forward on to the surface to be coated with a velocity in the region of 3000 feet per second. The pistol, it may be mentioned, weighs only about 3½ pounds.

It is essential that the surfaces to be sprayed should have an open texture, and with metallic surfaces this is obtained by sandblasting, while for wood, if the material is dry, either a light sand blasting should be given or if this is not convenient, the surface should be rubbed with sandpaper. If the surface is properly prepared, it is claimed that the sprayed metal becomes practically an integral part of the surface, which cannot be separated from it either by hammering or bending, there being thus no danger of peeling or flaking. Indeed, in most cases when attempts are made to remove the deposits, the underlying metal is fractured instead of the deposited metal being removed. Nickel is the metal usually employed for building up worn parts, since this yields a surface which can be easily machined or ground, and which, moreover, is sufficiently hard to give a high resistance to further

wear, while the deposits can be applied in thicknesses from one-thousandth up to one-eighth of an inch. For protection against corrosion zinc is employed, a sprayed zinc coating of 0.4 thousandth giving an equivalent protection to hot-dip galvanizing. For the protection of iron or steel against the effects of scaling due to exposure to high temperature, aluminum is employed, and it is of interest to note that the superheater elements of the *Bremen* have been protected with aluminum applied by a special aluminizing process developed by the Metal Coating Co. Ltd. Condenser tubes have also been protected internally by aluminum in this way, a rotary nozzle 5 feet long being employed. As a further example of the application of this process, * * * a screw on one of the Hamburg-Amerika liners, which had pitted. * * * The pittings were filled and the blades smoothed over with bronze. * * *—"Shipbuilding and Shipping Record," Nov. 7, 1929.

ACCUMULATORS FOR USE ON SHIPBOARD.

In a previous issue,* under this title, we published a description of the Young accumulator, manufactured by the Young Accumulator Co. (1929), Ltd., of London, and we are now returning to the subject in order to direct attention to a new improvement which has just been introduced by this progressive firm.

Before dealing with this, however, it may be an advantage to review the claims already made for the Young accumulator. By the use of special materials, sulphation of the positive plates, which is responsible for a decreased efficiency, is entirely eliminated, and the internal resistance is stated to be 50 per cent lower than that of other batteries. The increase in capacity for the same weight of electrodes varies from 20 per cent to 100 per cent, according to the rate of discharge, which may be greater than that usually permissible with other types. The charging rate may also be considerably increased without impairing the life of the battery or its efficiency. When it is necessary that the batteries should be unspillable, a solid electrolyte, known as "Dri-power," is used. This, in conjunction with the special plates, will ensure a capacity at least equal to that of any lead-acid battery, and at high rates of discharge a 50 per cent increase in capacity is claimed. In the article referred to, we published results of tests carried out at the National Physical Laboratory, Teddington, which substantiate these claims.

The latest improvement consists of the use of specially-treated pure-rubber separators in place of the more usual wooden ones. This innovation, which is known as the "Wilderman" separator, is claimed to have a porosity forty thousand times greater than any known method of separating the plates. By its adoption, the internal resistance of the Young accumulator, which was already extremely low, has been still further reduced to an almost infinitesimal quantity. The resultant efficiency is thereby extremely high, and the battery has an increased capacity which enables discharges to be effected at high rates. A further advantage is that there is no loss of current when on open circuit. The Wilderman separator is quite impervious to any chemical action, and is exceptionally light for the work which it performs.

Through the courtesy of the manufacturers, we visited their very well-equipped new works, which have recently been completed on the Kingston By-pass Road at New Malden, Surrey. On this occasion we had the pleasure of witnessing very convincing tests of their new separator. The exceptional

* No 219, Vol. XXXV., p. 679.

porosity was demonstrated by blowing smoke through a sheet of the material, which, when held in front of a strong light, was quite opaque. A further test consisted of throwing copious quantities of water over one of the sheets, when the water was absorbed as though the sheet had been made of blotting paper.

Such a battery, having special non-sulphating plates and the new Wilderman separators, together with the solid electrolyte "Dri-power," will be a distinct advantage for marine use over the older types of batteries, whose faults are only too well known. The catalogue published by the Young Accumulator Co. (1929), Ltd., already contains a number of useful appreciations from shipowners, together with a list of the users, which includes several of the important shipping lines.—"The Shipbuilder," December, 1929.

THE FÖTTINGER TORSIONMETER.

Much ingenuity has been expended in devices for transferring to a stationary indicator the readings of a torsion meter which necessarily rotates with its shaft. These are all dispensed with in the case of the simple pattern illustrated, the design of which is due to Dr. Föttinger. Here the indicator rotates with the shaft and its dial, being brightly illuminated by a properly shaded electric lamp which moves with it, there is, it is stated, no difficulty whatever in taking readings whether the shaft be running fast or slow. In the view reproduced in Figure 1, the torsion meter is shown in use in one of the mechanical laboratories at Charlottenburg, and the transmission shaft, complete with its sleeve, indicator and lamp, is shown to a larger scale in Figure 2. The arrangement of the mechanism will be easily understood on reference to Figure 3. The sleeve *a* is firmly secured at *b* to the transmission shaft, and, at its right-hand extremity, has mounted on it the disc *c*, which supports, as shown, the fulcrum of the bell crank *d*. The short end of this lever is connected by a link *e* with the back plate of the drum *f*,

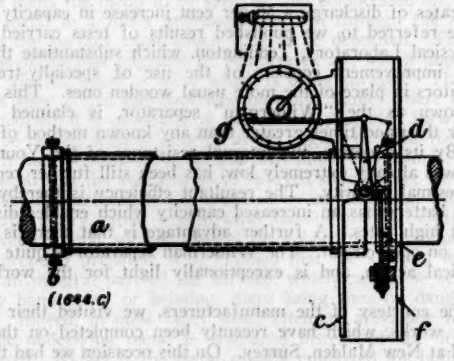

FIGURE 3.

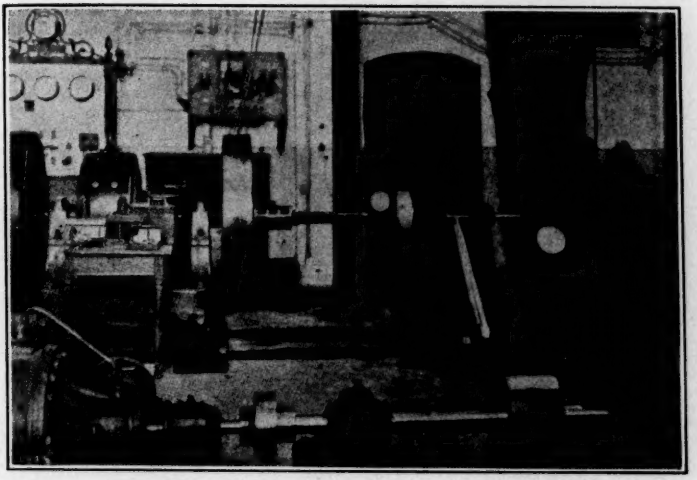

FIGURE 1.—TORSIONMETER IN USE ON TEST BENCH.

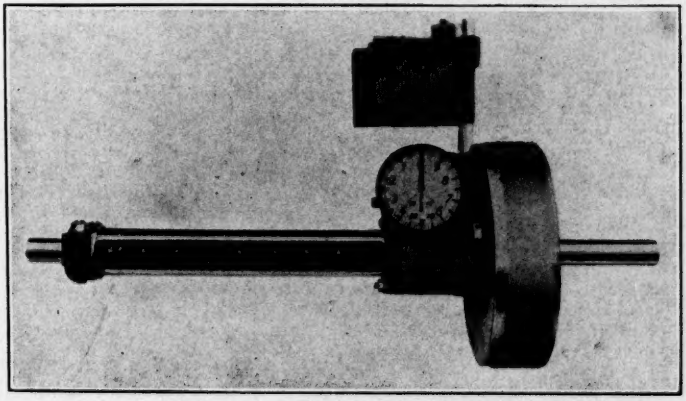

FIGURE 2.—TRANSMISSION SHAFT WITH TORSIONMETER IN PLACE.

which is firmly clamped to the shaft. When a torque is being transmitted, the cross section of the shaft at *b*, and with it the disc *c*, undergoes an angular displacement relatively to the drum *f*. By means of the link *e*, the bell crank *d* and the rack *g*, this displacement is shown, on a highly magnified scale, on the dial of the indicator. The observer stands so that the dial in question on the upper part of its path moves either directly to or directly from him, and under these conditions there is, it is stated, no difficulty in taking readings.

The sleeve, disc, and drum are all made of aluminum, so that even in the largest sizes the torsion meter is of moderate weight. It is made in nine standard sizes, for shafts ranging in diameter from 50 millimeters (1.97 inches) up to 625 millimeters (24.6 inches), and a series of interchangeable sleeves is provided by which the same instrument can be used for a range of shaft diameters. It is supplied by Messrs. The Lunken Company, Limited, 35, Great Dover-street, London, S.E.I.—"Engineering," Oct. 25, 1929.

RECORDING FAST OSCILLATIONS.

By H. Thoma.

V. D. I. 11 May 1929, p. 639. Translated by E. C. Magdeburger.

A new method to record, without distortion, mechanical oscillations electrically by means of a high frequency source and rectifying tube is described and a number of records given which were taken on different generator units driven by water and steam turbines, also on machine tools. These records show that in addition to the slow oscillations usually reproduced by instruments of the mechanically recording type there are many high frequency oscillations, which are important in diagnosing the cause of trouble. Additional spheres of usefulness in machine tool drives, in the building and testing of power machinery, in structural engineering, etc., are suggested.

Definition of the Problem.

Minute changes in length, such as for instance the elongation of structural girders, may be measured by transferring the motions involved to two parallel metal plates, installed with a small air gap, square to the axis of the motion and forming part of a high frequency circuit. The changes in the electrostatic capacity, which varies with changes in the air gap, affect the high frequency vibrations and result in changes of its electrical characteristics, such as current and voltage. The measurement of the latter will permit conclusions on the distance of the air gap and consequently the required elongation. Such arrangement is, however, not satisfactory for measurement of quickly changing phenomena, since particularly the instruments suitable for high frequency are usually very inert, and would follow the changes in the air gap but slowly. It was desirable to make such an arrangement available for faultless recording of even the fast changing phenomena. It was then necessary first of all to so strengthen the weak impulses of high frequency current, that it would become sufficient to operate fast recording instruments, and on the other side, see to it that the high frequency oscillations themselves would not act disturbingly. After a few experiments at the High Potential Laboratory of the Karlsruhe Institute of Technology, it became possible to embody these ideas into a comparatively simple apparatus. Since already the first applications show that by

these means more light can be thrown upon vibration problems, than with the older instruments which are mostly of the mechanical type, the fundamentals of the new method are going to be described herewith.

ARRANGEMENT AND METHOD OF USE.

The fundamentals of the required circuit are shown on Figure 1.

The left side of the circuit shows the essentials of a high frequency sending apparatus, consisting of a vacuum tube R_1, such as is commonly used in radio; this tube sets up vibrations in the circuit S_1 of the high frequency sender with the aide of plate battery AB_1. It is desirable to maintain this vibration at a somewhat constant rate of frequency during the experiment. Suitably loosely coupled to this oscillatory circuit, by means of a coupling coil K, for instance, is another resonant circuit S_2 which includes one condenser, regulated by hand, and another one which is marked C_2 on Figure 1. C_2 consists of the two measuring plates and the variations of the air gap between these two it is desired to record.

When the resonant circuit S_2 is so tuned to the oscillatory circuit S_1, or the sending frequency ν_1 of the high frequency sender, that they are close to resonance, then a very small change in the capacity C_2 results in substantial change in the current passing through the resonant circuit S_2. The relation between current and voltage and natural frequency ν of the circuit S_2 can be approximately represented as shown on Figure 2 schematically.

If the natural frequency is so chosen, that for the mean distance between the plates of the condenser C_2 it corresponds to about the value ν_1 then a minute change in capacity or, what is the same, in the distance between the plates of the condenser C_2 will result in a large change in current and voltage of the resonant circuit S_2. The steepness of the corresponding part of the curve increases with the decrease in damping chosen for the vibrating circuit, and in practice it is not difficult to keep this damping down to a very small figure and therefore it is quite easy to get a very steep resonance curve. It is also possible to choose the mean operating position so that it is near the turning point of the curve and a practical proportionality will exist between the minute changes in distance between the plates, or the air gap which is being measured, and the currents or voltages of the resonant circuit S_2.

Systems of such and similar character are often used for the measurement of small changes in length. In order to achieve rapid recording of the variations in the air gap between the plates and thereby changes in length, rectification becomes necessary. Besides it is usually desirable to include further amplification. This is done by means of the incandescent cathode or vacuum tube R_2 in Figure 1, whose regulating voltage is taken from the resonant circuit, by means of a suitable connection to the grid. This regulated voltage causes the sending out of direct current of variable density from the incandescent filaments of this tube, which are heated by means of the battery H_2. The direct current thus sent out is caught up by the anode A_2, as is customary in radio practice, and led through oscillograph O. The oscillograph, commonly used in electrotechnik, which is our case was of Siemens & Halske make, is essentially a string galvanometer with a photographic recording device. Its natural frequency may be without difficulty maintained at about 5,000 to 10,000 Hertz,* so that the use of this already available fast recording apparatus opens the immediate possibility of recording all oscillations and mechanical vibrations with frequencies up to that of the sound without any appreciable drag or retardation.

* Hertz equals one vibration per second or V. P. S.

Important for the faultless recording is, of course, the use of exciting frequency, considerably in excess of the natural frequency of the oscillograph. The experiments described below were made with a frequency of about 3 million Hertz. On the other hand it can be easily demonstrated that the unavoidable building up phenomena of the high frequency system produce no noticeable retardation by recording on the oscillograph the sudden switching on and off of small additional capacities. By a suitable choice of conditions it is possible to make it so that no other retardation will be effective except that resulting from the above mentioned natural frequency of the oscillograph itself, which is very much higher than any mechanical vibrations which may be investigated.

The method permits therefore the recording of mechanical oscillations and vibrations practically without distortion and phase retardation or inertia effects. Such records showed that vibrations usually consist of several very complicated and usually non-harmonic types of vibration superimposed one upon another. It may appear at first glance that the knowledge of such vibration components for machinery, structural systems and parts of ships hull is unnecessary, but experience showed that they have often helped in the diagnosis and the removal of the cause of vibration, which previously baffled other means of detection. Further improvements in the system, the description of which may lead too far at this time, have increased the sensitivity of the measurement so far above that possible with instruments of the mechanical type, that magnifications up to million to one become feasible for instance, and such a small change in the air gap as .000001 inch would be recorded as 1 inch on the diagram.

While experiments for the perfection of the method were in course of progress, a new method serving similar purpose has been described by Sachsenberg.* His method, however, permits recording of vibration curves of points with comparatively large time intervals, so that no real high speed recording is possible. Records produced by the above described new method are in themselves a sufficient proof of the desirability and importance of a continuous record of vibration phenomena, a record without distortion.

APPLICATIONS OF THE METHOD.

Figure 3 shows, for instance, the vertical movements of the center bearing of a large water turbine driven generator unit, the shafting of which is represented schematically on Figure 4. Figure 3 shows that the center bearing vibrates largely in synchronism with the third harmonic, *i.e.*, makes three vibrations per revolution. This corresponds exactly to the mode of vibration shown on Figure 5, whose frequency was estimated by the builders and the author at about 1600 V.P.M. (vibrations per minute). This confirms the probably not very well known law that *it is dangerous to operate generating units at speeds equal to any whole number of their frequency.*

Föppl and Stodola have given as a possible reason for these oscillations the periodic changes in torque. These reasons which are numerically very insignificant as a rule, could not have been the cause here, since the oscillations did not disappear even during no-load operation. Irregularities of the magnetic field, of the armature or of the air gap had to be excluded also as possible reasons, since oscillations of similar form but of slightly smaller amplitude showed up even for widely varying conditions of excitation. If the oscillations were smaller here, the reason for it was the not generally considered fact that with the excited field the natural frequency of the armature and shaft is lowered on account of the reaction of the magnetic forces,

* V.D.I. (1927), p. 1609 and (1928), p. 469.

and thus approaches in this case a value of three times the shaft speed. The reason for the oscillations was discovered finally, after not only the existence of these oscillations was ascertained with the aid of the oscillograph, but also their phase relation. It was found that the two keys located 120 degrees from each other on the hub of the centrally cored out turbine wheel have periodically influenced the shaft distortion and synchronized with its natural frequency. Superimposed upon this oscillation, which repeats itself three times per revolution, a row of smaller and faster oscillations is noticeable in Figure 3. These are, as calculations have shown, the result of the bucket impulses of the Pelton wheel type of turbine. Figure 7 shows the arrangement of the air gap condenser used for recording vertical vibration of the bearing pedestal, and Figure 8 shows the general arrangement of the apparatus. In addition to recording vertical vibration of the bearing pedestal, vertical vibration of the shaft itself was also recorded (See Figure 9), using for that purpose a pulley adjacent to the bearing. Here one of the condenser plates was pressed against the polished surface of the pulley with the aid of a slipping spring, while the other condenser plate was held as before in a heavy U-formed stand, resting on both sides of the machinery directly on the foundation. The pulley was slightly eccentric and a slight wave per revolution is visible on the record.

Remarkable are the records of passing through the two critical speeds of the shaft system, the first critical corresponding to the resonance of the bucket frequency of the Pelton wheel with the natural frequency of the shafting as per Figure 5 and the second critical shown on Figure 9, resulting from the resonance of the bucket frequency with the first overtone according to Figure 6. Figure 10 reproduces the entire starting up to a correspondingly smaller scale, otherwise recorded as in Figure 9, and very clearly the rising of the wave, resulting from the appearance of the oil film between journal and bearing shell. This method of measuring the thickness of oil film is not only much simpler than others, but it shows also the details of how it develops. The natural frequency of the armature and shaft was determined also on the unit standing still, by fastening of the condenser plates to the stator of the generator and by setting the latter into vibration by a moderate hammer blow. Figure 11 shows the resulting record of vibration with the natural frequency of the unit as 1620 vibrations per minute and in different directions it proved to be different. That the asymmetry of the rotor thus shown cannot be a result of the bearing only, is proved by Figure 12, which records the easy mobility of the spherical bearing shells in the housing of the bearing, since here one of the condenser plates was fastened to one end of the bearing shell while the other plate rested on the pedestal itself.

Figure 13 shows the vertical vibration of the center bearing on a similar unit where the third harmonic is, however, only very slight, while other impulses predominate. These latter are caused partially by movements of the foundation, as can also be seen in Figure 11, to the left, since the heavy and comparatively slow running shaft of the unit, as in a seismograph, permits the recognition of the relative motion between the foundation and the shaft as long as its natural frequency is not brought into play by an exterior force, such as was the hammer blow in our case for instance. Figure 14 shows a similar oscillogram of an entirely different unit, namely, a steam turbine alternator of 12,500 K.V.A. and 1500 R.P.M., having a rigid coupling with the shaft supported on both sides of it by a common pedestal. Vertical movements of the shaft on the turbine side of the bearing with reference to the pedestal itself were recorded at the normal speed of the unit without

load or excitation on the generator. The record shows a quick raising of the shaft every second revolution, while during the intermediate revolution the shaft moves lower down. The reason for it could be the periodical changes in the oil film thickness due to insufficient clearance in the bearing as well as bending of shaft from local overheating.*

FURTHER PRACTICAL APPLICATIONS.

It is evident that the reasons for vibratory disturbances can be studied and satisfactorily explained if a continuous record of the phenomena with an apparatus, not subject to inertia effects may be obtained. A way to correct them can usually be found as the result.

It is interesting to note that even machinery evidencing but little effect of periodic disturbance show a great variety of oscillations, each of which can be explained by some peculiar characteristic of that particular unit. Railway power plants, for instance, show the 33 periodic changes of torque characteristic of the 33 cycle single phase alternators transmitted to the foundation as shown by its minute movements. Overtones of this vibration are also usually easily detectable in these plants. The same applies also to electric light power plants with slightly unsymmetrical load, with this difference, however, that the lowest observable frequency here amounts to 100 per second for 50-cycle units.

With the continuously growing size of plants and individual units grows the danger that some periodic impulse originating in the generator units will synchronize with the natural frequency of some constructional detail of the plant. The complete record of oscillations, even of units giving no reason for objections, shows clearly how many of such potential dangers exist and how difficult it is sometimes to avoid them. Since amplitudes can be measured with precision up to .01 millimeter or even .001 millimeter, it is possible to avoid future trouble by specifying permissible amplitude values and ranges of natural frequency for the machinery.

The process described is also very suitable for recording deformation of parts of ships while underway including the fast oscillations caused by the propelling plant and by the action of the sea waves. Compared with the method based on recording the changes in resistance of a carbon-pile which we also used for our first tests, the above-described method has not only the advantage of greater sensitivity so that very short measuring lengths are sufficient, but also the absence of after effects, which unfortunately seem to appear constantly with the carbon-piles as our own tests have shown.

These after effects render it impossible to interpret the changes in the coal pile resistance in terms of movements of the constructional details, which have been measured, because the curves so recorded seem to show rather extensive distortions as was shown by comparison with records made by the oscillograph.

Another field of application is the testing of machine tools. The importance of measuring the forces and deflections on the work and the tool itself has been pointed out already by Schlessinger. He has, however, measured only undefined average values of these with his very inert apparatus. A clear conception of the actual processes on the machine tool is possible only through study of continuous records of the oscillations set up at the time and it is also quite certain that the size and kind of the oscillations are just as important to the quality of the work as to the life of the tools.

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* V. D. I., 1925, p. 985.

Figure 15 shows an example of the vertical movements of the support of a 10-inch lathe while turning axially a cast iron piece of metal of 2-inch outer diameter with a chip of 5 millimeters² cross-section. The significance of these for the first time clearly recorded vibrations for the construction and value of machine-tools and their tools is being investigated by Kessner and the author in the machine tool laboratory of the technical high school at Karlsruhe.

At the same time tests are in progress on high speed reciprocating engines. The great sensitivity and precision of the described method permit faultless recording of deflection of such very stiff parts as a valve disk for instance, which follows pressure practically without any lag, as well as recording at very high speeds which was utterly impossible with devices of mechanical type.

SUMMARY.

By means of two small metal plates movable with respect to each other a high frequency vacuum tube transmitter may be so influenced in a suitable circuit that very slight relative movements (less than μ) of the plates will produce considerable changes of the current and voltage. When the frequency of the transmitter is chosen high enough, say about several millions of vibrations per second, and when its impulses are rectified through a second tube with simultaneous amplification, then a direct current results that can be recorded by means of an oscillograph. The interruptions of this direct current corresponding to the high frequency alterations disappear for all practical mechanical purposes, so that the fast recording of the distance between the above mentioned measuring condenser plates is possible without distortion for all practical frequency ranges within the range of the oscillograph. Since the normal types of oscillographs have a natural frequency of 1000–10,000 vibrations per second all oscillations, shocks and length changes up to the highest frequencies, that are usually encountered in many mechanical problems, up to the frequency of the sound, can be recorded practically without any distortion. As contrasted with the formerly used means this method gives uninterrupted records of any vibration and without any distortions. The calibration of the amplitudes recorded by the oscillograph is done in the simplest manner through attaching a micrometer, which produces given displacements of the condenser plates for comparison with the records of the oscillograph.

The method has considerable importance for the investigation of the deformation of parts of ship's hull, structural details of buildings and bridges, machinery, and for the measurement of oscillations appearing in them. From such records a very clear conception of the phenomena of passing through the critical speeds is obtained. The method is also suitable for obtaining clear numerical data on the amplitude and frequency of vibration during builders' tests for machinery structures or vehicles.

N. B.—A further article by the same author, illustrating the application of the above described method to the problems of torsional vibration, which was responsible for failures of four propelling plants of Graf Zeppelin on her second attempt to cross the Atlantic, will be found in V. D. I. of 28 Septemper, 1929, on page 1383.

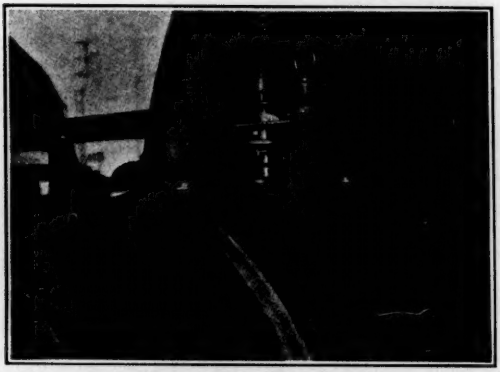

FIGURE 7.

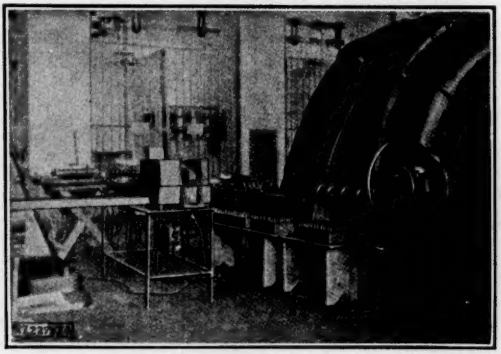

FIGURE 8.

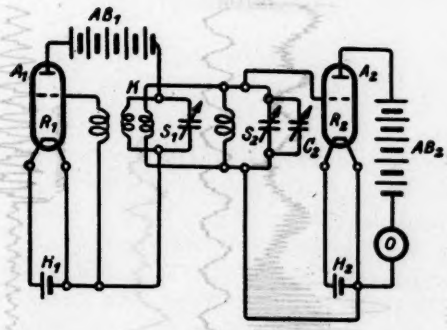

FIGURE 1.

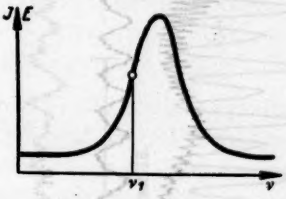

FIGURE 2.

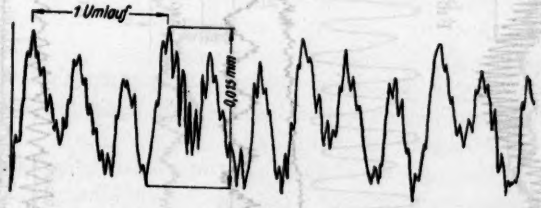

FIGURE 3.

FIGURE 4.

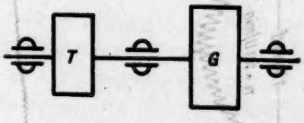

FIGURE 5.

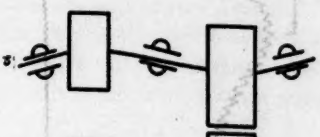

FIGURE 6.

FIGURE 9.

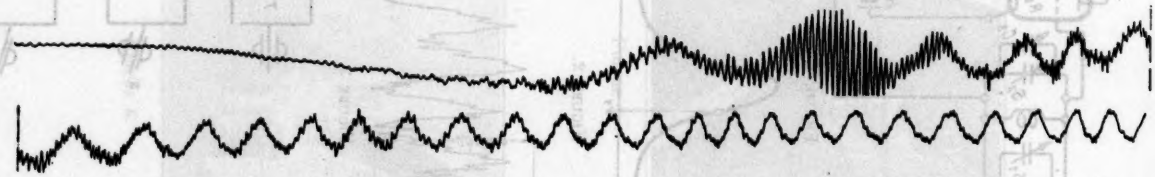

FIGURE 10.

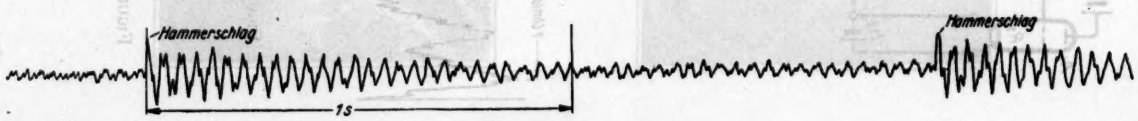

FIGURE 11.

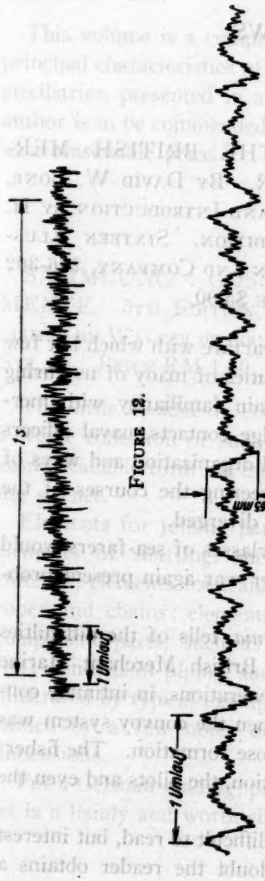

FIGURE 12.

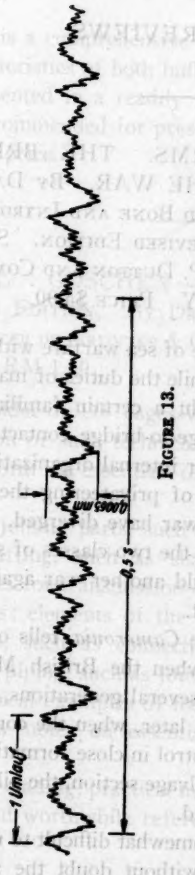

FIGURE 13.

FIGURE 14.

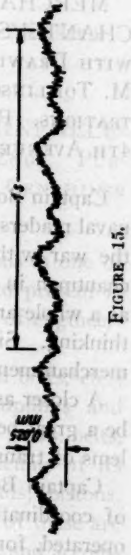

FIGURE 15.

BOOK REVIEWS.

MERCHANTMEN-AT-ARMS. THE BRITISH MERCHANTS' SERVICE IN THE WAR. By David W. Bone, with Drawings by Muirhead Bone and Introduction by H. M. Tomlinson; New and Revised Edition. Sixteen Illustrations. Published by E. P. Dutton and Company, 286-302 4th Avenue, New York, N. Y. Price $5.00.

Captain Bone writes of a side of sea warfare with which but few naval readers are familiar. While the duties of many of us during the war with Germany brought a certain familiarity with merchantmen in the form of bridge-to-bridge contacts, naval officers as a whole are ignorant of their internal organization and ways of thinking. Since the abolition of privateering the courses of the merchantmen and the man-of-war have diverged.

A closer association between the two classes of sea-farers would be a great benefit to each should another war again present problems of trans-oceanic transport.

Captain Bone, master of the *Cameronia*, tells of the difficulties of coordination encountered when the British Merchant Marine operated, for the first time in several generations, in intimate contact with the Royal Navy, and later, when the convoy system was adopted, under their actual control in close formation. The fishermen, the mine-sweepers, the salvage section, the pilots and even the light-ship services are discussed.

The book may at times be somewhat difficult to read, but interest is sustained throughout and without doubt the reader obtains a clear conception of the difficulties and often the hardships of war when viewed from the bridge of a merchantman-at-arms.

Sherwood Picking,
Lieutenant Commander, U. S. Navy.

MOTORSHIPS OF THE WORLD. By A. C. Hardy. Published by Sampson Low, Marston & Co., Ltd., London. Price 12/6 net.

This volume is a comprehensive collection of data covering the principal characteristics of both hull and engine including the main auxiliaries, presented in a readily readable tabulated form. The author is to be commended for presenting such a reference volume at a reasonable price.

James S. Malseed.

SAMMLUNG GOSCHEN—DIE MASCHINENELEMENTE. 5th Edition. By Dr. Ing. E. Vom Ende. Published by Walter de Gruyter & Co., Berlin W 10 Genthiner Str. 38. Price RM 1.50.

This little booklet of 118 pages with 152 illustrations is one of a whole collection of such technical books intended to present in easily readable form the essentials of various chapters of engineering science.

Elements for joining parts, such as rivets, keys and bolts, etc.; elements of shafting, such as shafts, bearings, couplings and clutches; elements for transmission of energy, such as gears, belts, ropes and chains; elements of the crank gear and details of its component parts, such as connecting rods, crossheads, pistons, etc.; elements of piping, such as joints and valves—all of these are illustrated by typical examples of modern construction and accompanied by a few notes on essentials of calculating their basic dimensions.

For a German-reading, practical mechanic this inexpensive booklet is a handy and worthwhile reference book to consult.

E. C. Magdeburger.

ASSOCIATION NOTES.

ELECTION OF OFFICERS.

Ballots cast for officers of the Society for 1930 were counted at a meeting of the Council held in Washington, D. C., on Wednesday, January 8, 1930, and the following were declared elected:

President:

Rear Admiral H. E. Yarnell, U. S. Navy.

Secretary-Treasurer:

Commander H. T. Smith, U. S. Navy.

Council:

Rear Admiral C. W. Dyson, U. S. Navy, Retired.
Captain I. E. Bass, U. S. Navy.
Captain O. L. Cox, U. S. Navy.
Captain H. S. Howard (CC), U. S. Navy.
Captain R. B. Adams, U. S. Coast Guard.
Mr. H. M. Southgate.
Mr. J. F. Metten.

ANNUAL BANQUET.

The annual banquet of the Society will be held at the New Willard Hotel, Washington, D. C., on Friday, May 9, 1930. Full information relative to this annual event will be furnished the membership in the near future.

PRIZE ARTICLE, 1929.

"Theoretical and Experimental Study of Condenser Scoops" by Mr. Henry F. Schmidt has been chosen by the Council as the Naval Engineers Prize Essay for 1929. The paper is published elsewhere in the current number of the JOURNAL. This is the second prize essay written by Mr. Schmidt, the previous one, "Some Screw Propeller Experiments," being the Prize Essay for 1927.

"A Plea for Sounder Engineering Competition" by Lieutenant-Commander Roger W. Paine has been awarded honorable mention for 1929.

FINANCIAL STATEMENT.

Following is a financial statement for the calendar year 1929:

RECEIPTS AND EXPENDITURES.

Income.

Publication:			
Dues		$5,840.00	
Subscriptions		2,765.65	
Sales		738.16	
Advertisements		3,943.50	
Exchange		.66	$13,287.97
Interest on Investments and Bank Account			872.38 $14,160.35

Expenditures.

Publication:			
Printing		$8,152.44	
Engraving		1,386.54	
Manuscript		845.00	
Postage		157.52	
Commission and Discount		229.69	$10,771.19
Salaries			2,700.00
General Expense			137.11
Current Profit and Loss			368.29
Banquet, 1929			79.07 14,055.66
Net Gain			$104.69

BALANCE STATEMENT.

Assets December 31, 1929:			
Cash			$3,114.43
Accounts Receivable:			
Advertisements		$1,599.13	
Dues		375.75	
Sales		85.99	
Subscriptions		21.75	
Miscellaneous		6.25	
Certified Check to Navy Department in lieu of Bond		271.25	2,360.12

Investments (all Bonds)		
Armour and Co.	$887.50	
Army and Navy Club	1,000.00	
Bethlehem Steel Co.	957.50	
Canadian Northern Railway	1,115.00	
Hocking Valley Railway	1,682.50	
International Mercantile Marine	795.00	
New York Steam Corporation	927.50	
Ohio Power Company	860.00	
Pennsylvania Power & Light Co.	882.50	
Washington Gas Light	997.50	
Washington Ry. and Elec. Co.	4,230.00	14,335.00
Furniture		197.25
		$20,006.80
Liabilities:		
Dues Paid in Advance	$125.50	
Subscriptions Paid in Advance	880.85	1,006.35
Net Assets		$19,000.45
Assets January 1, 1929		18,895.76
Net Gain		$104.69

The President has appointed a committee composed of Captain H. R. Greenlee, U. S. N., Lieutenant Commander J. K. Esler, U. S. N., and Lieutenant Commander B. R. Alexander, U. S. N., to audit the accounts of the Secretary-Treasurer as submitted.

MEMBERSHIP.

The following members have joined the Society since the publication of the November, 1929, JOURNAL:

NAVAL MEMBERS.

Balderson, James R., Boatswain, U. S. C. G.

Boyle, Charles L., Lieutenant, U. S. N.

Brooks, Winfield A., Lieutenant, U. S. N.

Dorward, Joseph E., Lieutenant, U. S. N. R., 600 W. 116th St., New York, N. Y.

Huse, John O., Lieutenant, U. S. N.

Lull, Edward E., Ensign, U. S. N.
Newhaus, Herbert M., Ensign, U. S. N.
Taylor, Edwin J., Lieutenant, U. S. N.

CIVIL MEMBER.

Blackwood, William James, 6315 Maple St., Chevy Chase, Md.

ASSOCIATE MEMBERS.

Caley, W. T., Westinghouse Electric and Manufacturing Co., 10 High St., Boston, Mass.
Colotto, Federico, Ansaldo Shipyard, Genoa, Sestri, Italy.
Fisher, C. R., Hollenden Hotel, Cleveland, Ohio.
Nicholas, Henry M., Westinghouse Electric and Manufacturing Co., Lester, Pa.

ERRATA SHEET.

Article entitled "Theoretical and Experimental Study of Condenser Scoops" by H. F. Schmidt in the February, 1930, (Vol. XLII, No. 1) JOURNAL:

Page 3, 6th line, change $\frac{A_1 V_1}{V_2}$ to $\frac{A_1 V_1}{A_2}$

Pages 25 and 26, replace Figures 30 and 31 with Figures herewith.

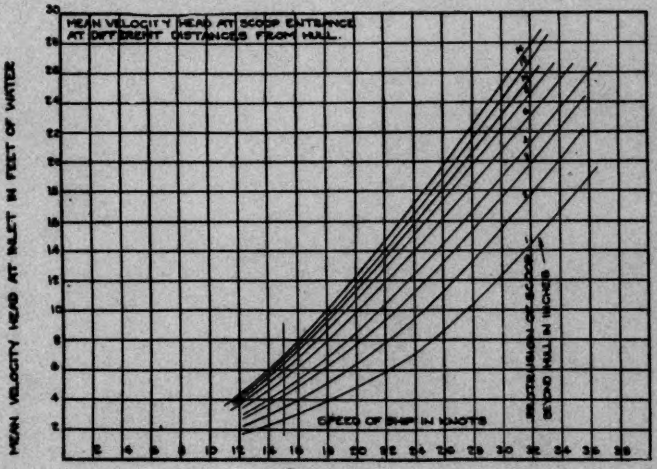
MEAN VELOCITY HEAD AT SCOOP ENTRANCE AT DIFFERENT DISTANCES FROM HULL.
MEAN VELOCITY HEAD AT INLET IN FEET OF WATER
SPEED OF SHIP IN KNOTS
DISTANCE OF SCOOP BEYOND HULL IN INCHES

FIGURE 30.

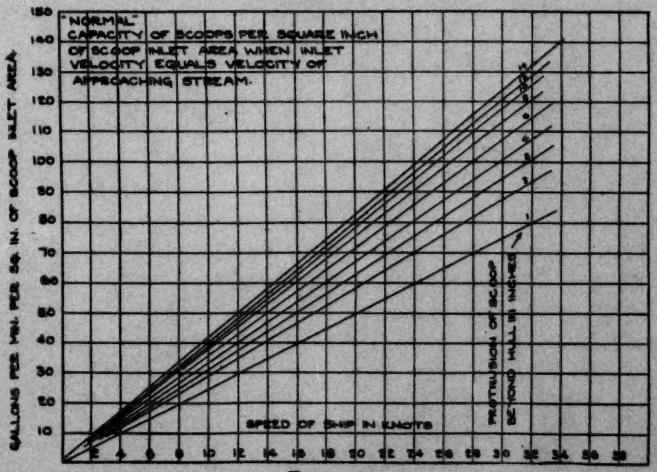
"NORMAL" CAPACITY OF SCOOPS PER SQUARE INCH OF SCOOP INLET AREA WHEN INLET VELOCITY EQUALS VELOCITY OF APPROACHING STREAM.
GALLONS PER MIN. PER SQ. IN. OF SCOOP INLET AREA
SPEED OF SHIP IN KNOTS
DISTANCE OF SCOOP BEYOND HULL IN INCHES

FIGURE 31.